THE INDICATOR DIAGRAM.

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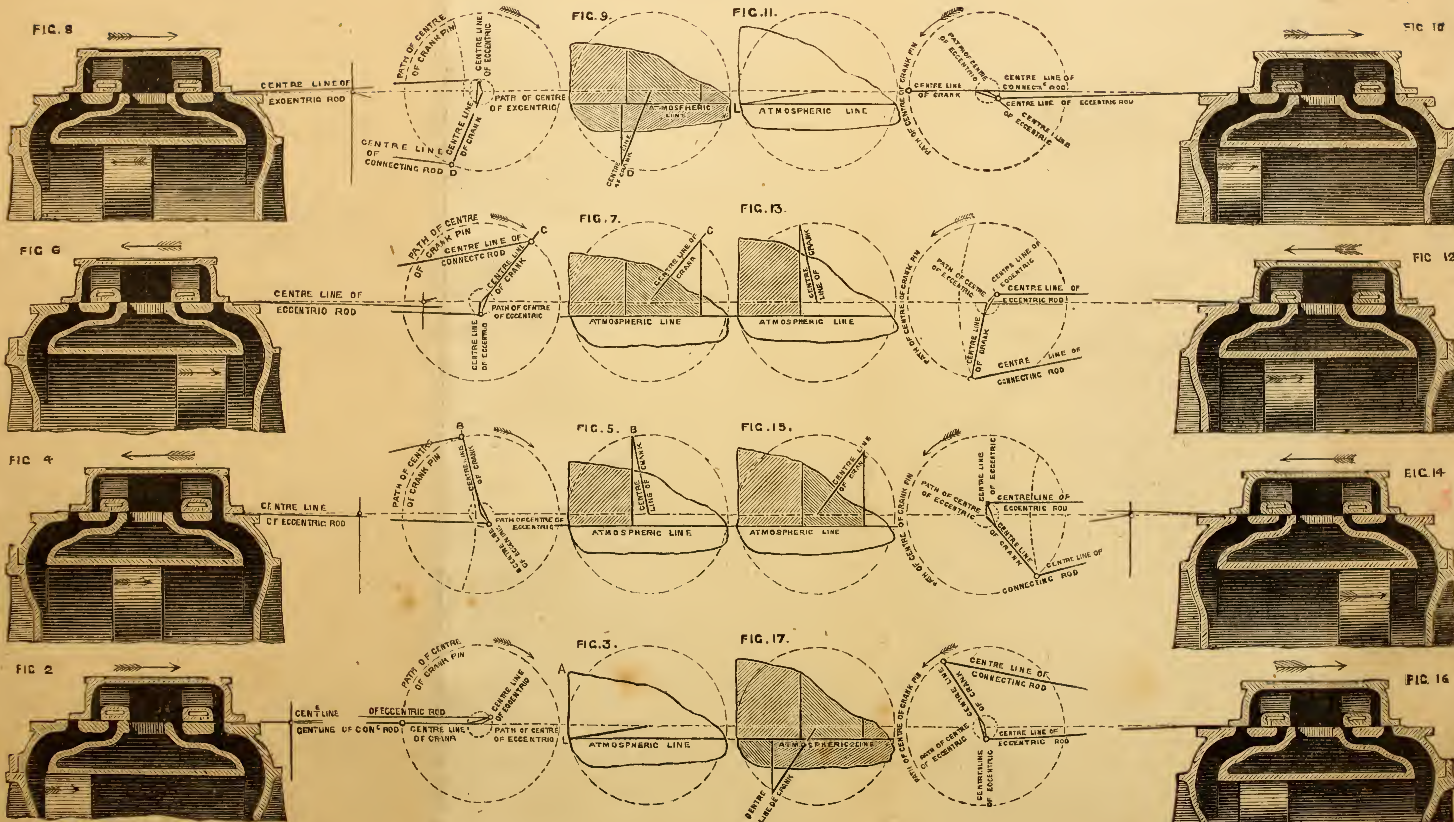
PRACTICAL GEOMETRICAL TESTS OF INDICATOR DIAGRAMS.

By N. P. Burgh, M.E.



Frontispiece.

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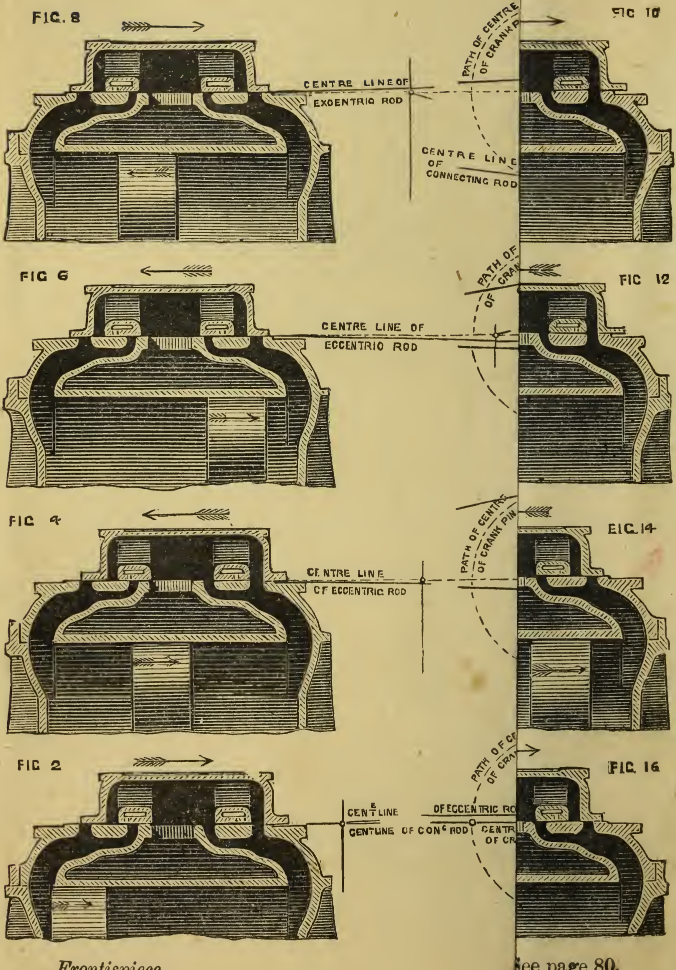

Frontispiece.

See page 80.

THE

INDICATOR DIAGRAM

PRACTICALLY CONSIDERED.

BY

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PREFACE.

This work was actually commenced in the latter end of the year 1867, when I had occasion to professionally test some indicator diagrams. The means I adopted were to consider, first, the proportions of the details that regulated the admission, expansion, and exhaustion of the steam, and secondly, their geometrical delineation. Having proceeded thus far, I then concluded that as the length of the motion for the indicator barrel was virtually that of the stroke of the piston, the scale of the geometrical test that I employed must be subject to that circumstance. I accordingly made a drawing of the cylinder ports, slide-valve, link motion, eccentrics and rods, a portion of the piston, length of the connecting rod, and

crank circle at the same scale that the stroke of the piston in inches was equal to the length of the indicator diagram. I acquired thus a correct knowledge of the positions of all the details that regulated the steam at the points of admission, cut-off, expansion, exhaustion, compression, and lead, in relation to the positions of the piston and crank pin at the same scale as the diagram; and by laying it on the crank-pin's circle I saw at once *how it was formed in relation to time and speed*, and the *cause of the defects*, if any existed; indeed, being the only method of gaining that information *truthfully*, which is illustrated by the plate as the frontispiece.

After I had well succeeded with my test, I looked around me to see if any one else had published a similar process, but could not find any record, or even a hint on the subject. I therefore thought it worth while to put the matter in print for public use, intending at that time to make a small work only on that special branch. On conversing, however, with a few of my professional friends on the affair, they urged

me not to condense the matter thus, but to make a *really practical* work on the *entire* subject, as it was much needed, at the same time offering me all the assistance necessary for the purpose: and when I mention the names of Messrs. Penn, Maudslay, Watt, Rennie, and Napier, I feel no hesitation in stating that no author but myself has been so honoured with their contributions, not only in this case, but in others when I have solicited information from them. I am therefore enabled to put forth the most reliable information from those firms on the indicator diagram in ten chapters, in the following order.

Chapter I. contains the description and use of the indicator, illustrating, at a working scale, Messrs. Maudslays' and Mr. Richards' Indicators.

Chapter II. is under the heading, "How to take an indicator diagram correctly:" which treats of the action of the steam in the cylinder; the definition of the diagram; correct indicator gear for horizontal, vertical, and oscillating engines, and indicating notes.

Chapter III. deals with the proof of atmo-

spheric pressure and particulars of steam pressures, and includes rules and tables bearing on the subject, with practical examples and illustrations.

Chapter IV. is a complete description and illustration of the theoretical geometry of the indicator diagram in a more practical manner than hitherto published.

Chapter V. is the practical geometry of the indicator diagram, which fully explains the frontispiece plate and other figures in connection with it.

Chapter VI. commences the illustrations of indicator diagrams and gear constructed by the firms I have mentioned, and others whose names are noted under the figures: this chapter contains twenty-one diagrams from ordinary modern screw engines of all classes; three from compound engines; three of steam-launch engines; and two illustrations of indicator gear for return acting engines.

Chapter VII. is devoted to diagrams taken from the most modern paddle-engines, and con-

tains eighteen illustrations of them; also the most improved indicator gear by Messrs. Penn and Napier, for oscillating engines.

Chapter VIII. treats of land-engine indicator diagrams, showing eleven examples taken from various classes, including locomotive engines.

Chapter IX. fully explains and illustrates air and water pump diagrams.

Chapter X. closes this work, with the explanation of the indicated horse-power in connection with the diagram.

The whole of this is expressed in one hundred and sixty-four pages, and illustrated by one hundred engravings; so that the entire matter has been fully explained, but not more so than requisite.

N. P. BURGH.

78, *Waterloo Bridge, London,*
January 1, 1869.

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THE INDICATOR DIAGRAM

PRACTICALLY CONSIDERED.

CHAPTER I.

THE DESCRIPTION AND USE OF THE INDICATOR.

A very large proportion of the young members of the engineering profession look at an indicator diagram as a mysterious production; and even supposing that they comprehend how it is formed, they do not understand the causes for the various forms of figures. There is, therefore undoubtedly, room for a practical work on the subject which shall deal with the matter just as a learner requires: leading him on step by step without slipping, and impressing on his mind all the realities of the case; for directly an author skips over any portion of his subject, with the phrase appended, "this can be readily understood," it is generally assumed that he does

not know it himself, and shuffles out of the argument with bad grace. The best course of instruction for the young engineer then is, that the truth of each minute portion of the subject before him be laid bare, so that he can comprehend it. When he understands thus far, obviously his knowledge will enable him to put the same into practice boldly: for by being acquainted with the ground-work of the subject, he is master also to a great extent of the result of his labours.

First let us consider with what we have to deal—it is the steam cylinder of an engine, with its piston in motion, and our aim is to explain what the steam is doing, or how the piston is impelled from the commencement to the termination of its stroke. The next question is, how can we effect this? Simply, by an instrument termed an "indicator," which is a cylinder fitted with a piston, surmounted by a spring to keep the piston always down to its work, when in contact with the steam. The piston-rod is in connection with a pencil, or marker, which indicates the motion of the piston up or down. The length of the stroke of the engine piston, of course, exceeds that of the indicator diagram in practice, while the principles of both are the same. The paper

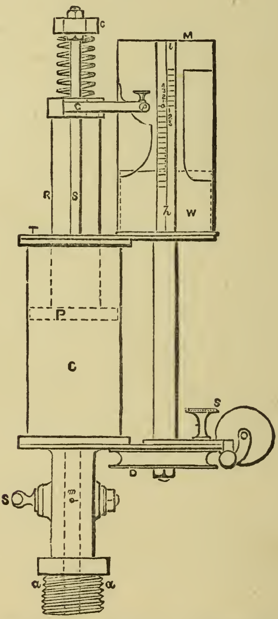
on which the diagram is traced can be laid flat if preferred during the operation, but generally it is wound around a barrel, which receives its motion—nearly rotary—from the piston-rod of the engine. There is, however, a certain discrepancy in the present mode of indicating which has been overlooked; it is this—the indicator piston is always exposed to the atmospheric pressure, while the engine piston is often closed from it.

This is the fault with all steel spring indicators of the present day, which not only permit the atmosphere to act on the indicating piston, but also require a spring to ensure that the piston follows the decrease of the pressure of the steam. A *perfect* indicator, for example, is that which is arranged so that its piston and the engine piston are operated on equally by the same volume of steam, proportionately to their areas. The indicator piston should be *weighted* in proportion to the inertia the engine piston has to overcome, so that the resistance in each case shall be alike relatively.

A good form of direct-acting indicator is shown by Fig. 1, as constructed and used by Messrs. Maudslay, Sons, and Field, the author being indebted to Joshua Field, Esq., for the drawing. Its combinations and arrangement are as follow:

C is the steam cylinder open at the top T. The piston P is fitted steam-tight, and transmits its motion by a tube R, the top of which is guided

Fig. 1



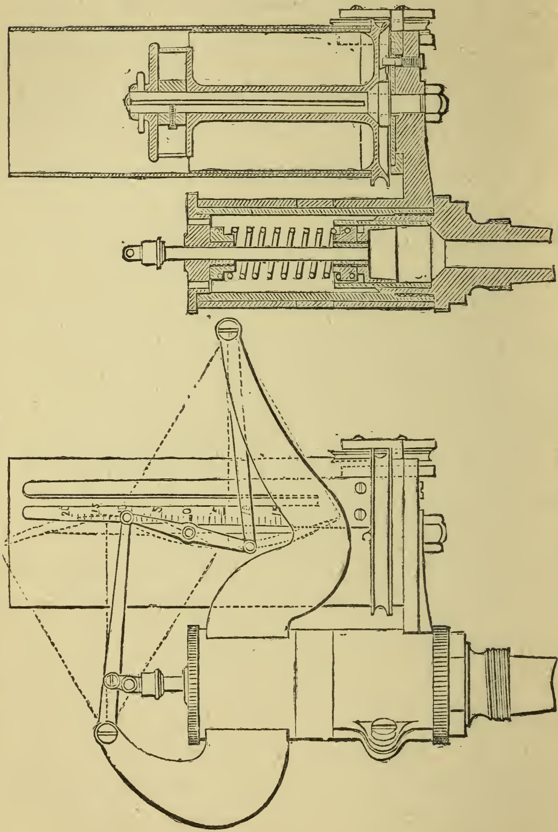
Messrs. Maudslay's Indicator. Scale quarter size.

by side rods S. The tube R contains a spring, which is prolonged, as shown, to the cross-piece *c* at the extremities of the side rods S. The marker or pointer *p* is fixed to an arm G, pro-

jecting from the guide at the top of the tube R, so that the piston P and pointer *p* move simultaneously. The motion barrel M, on which the paper is wound and held by the vertical spring *h*, is also fitted internally with a watch-like main-spring W, to ensure that the barrel will always tend to resume its original position when in action. The barrel is mounted on a rod, which is supported at the side of the base of the cylinder C, below which is the main pulley D, and the guide pulley G, the position of the latter being fixed as required by the set screw *s*. The steam is admitted and shut off by the plug-cock S, and the clearance of the condensed steam is effected through the little opening *m*. The screw *a a* at the extremity makes the connecting-joint either with the engine cylinder cover direct or with the branch pipe, as required.

Another kind of indicator is constructed by Messrs. Elliott Brothers, London, and illustrated by Fig. 2. Its principal object is to reduce the motion of the piston as much as possible, which is attained by the ordinary parallel motion shown in the side elevation, where the pencil or marker is depicted in the centre of the connecting bar. This arrangement is evidently entirely foreign to the original indicator, with which the height

Fig. 2.



Mr. Richards' Indicator. Scale $\frac{3}{8}$ inch = 1 inch.

of the diagram taken, and the stroke of the indicator piston, were alike in dimensions, where, as in the present example, the pencil travels about 3·5 times of the motion of the piston. This

difference in the two motions *magnifies* the action of the piston so much, that the pencil indicates the least convulsive movement of the piston, as well its direct action up and down, caused by the steam and the spring. The makers' description of their indicator is this:—

"The indicators are made of a uniform size, the area of the cylinder is one-half of a square inch, its diameter being ·7979 of an inch. The piston is not fitted quite steam-tight, but is permitted to leak a little; this renders its action more nearly frictionless, and does not at all affect the pressure on either side of it. The motion of the piston is $\frac{25}{32}$ of an inch, and the motion of the pencil, or extreme height of the diagram, is $3\frac{1}{8}$ inches. The paper cylinder is 2 inches in diameter, and the length of the diagram may be $5\frac{1}{4}$ inches if this extent of motion is given to the cord. The diagram is drawn by a pointed brass wire on metallic paper. This is a great improvement over the pencil; the point lasts a long time, cannot be broken off, and is readily sharpened, and the diagram is indelible. The steam-passage has two or three times the area usually given to it. The stem of the indicator is conical, and fits in a corresponding seat in the stop-cock, where it is held by a peculiar coupling, which is a circular

nut with two screws of unequal pitches. This arrangement permits the indicator to be turned round, so as to stand in any desired position, when, the coupling being turned forward, the difference in the pitches of the screws draws the cone firmly into its seat; and when the coupling is turned backward, the cone is by the same means started from its seat. The leading pulleys may be turned by some pressure, to give any desired direction to the cord, and will remain where they are set. By these means the indicator can be readily attached in almost any situation.

"*The Springs.*—In order to adapt this indicator for use on engines of every class, springs are made for it to nine different scales, as follows:—

No. 1, $\frac{1}{8}$-in. motion, shows 1 lb. pressure on the sq. in. Indicates from .					—15 to + 10
" 2, $\frac{1}{12}$	"	"	"	"	—15 " + 22·5
" 3, $\frac{1}{16}$	"	"	"	"	—15 " + 35
" 4, $\frac{1}{24}$	"	"	"	"	—16 " + 60
" 5, $\frac{1}{24}$	"	"	"	Atmosphere	to + 75
" 6, $\frac{1}{32}$	"	"	"	"	" +100
" 7, $\frac{1}{40}$	"	"	"	"	" +125
" 8, $\frac{1}{48}$	"	"	"	"	" +150
" 9, $\frac{1}{56}$	"	"	"	"	" +175

"All the scales, except No. 2, are multiples

of 8, and the common rule will measure all the diagrams, if the proper scale is not at hand. It will be observed that the five higher scales do not indicate the vacuum. These are so made for the following reasons : the far greater number of engines which work steam at high pressures do not condense, and moreover, at these pressures, the scale of the indication necessarily becomes small, while it is always highly desirable to show the vacuum on a large scale. Spring No. 1 may be employed to indicate the vacuum, in engines which work steam at high pressures and with condensation. It can be readily substituted in the indicator, and the diagram which it will give will be on a satisfactory scale. It is provided with a stop, which prevents it from being compressed too much, so that a high pressure of steam will not injure it. Moreover the vacuum being omitted from the scales which go above 60 lbs., the entire range of the pencil is available for the pressures above the atmosphere, which are therefore shown on a somewhat larger scale. The springs indicating pressures above 60 lbs. will be made, however, to indicate the vacuum also, when so ordered."

The next consideration is the scale of the diagram, for if the pressure of the steam is

increased in one example to that of another, and the heights of the diagrams are alike, obviously the scale of each cannot be the same. For example, if $2\frac{1}{2}$ inches equals the height of the diagram, or of the upward motion of the indicator pencil from the atmospheric line, and the scale is equal to $\frac{1}{16}$th of an inch per pound of steam on the square inch, then $2{\cdot}5 \times 16 = 40$ lbs. on the square inch; but if the scale is $\frac{1}{20}$th, then $2{\cdot}5 \times 20 = 50$ lbs. We learn from this that the piston of the indicator, if unaltered in its area, must be loaded relatively to the pressure of the steam, and the scale of the diagram must correspond also.

It will be noticed that, for their indicator, the Messrs. Elliott adopt nine different springs, each to correspond with certain pressures and scales, the stroke of the piston being alike in all cases. The result, then, is nearly the same as would be attained by proportioning the area and load on the piston to the pressure of the steam in the engine cylinder.

A really *perfect* indicator can be made, however, by arranging the detail so that the steam takes the place of the steel spring or load over the piston; by which arrangement the load can be increased or decreased to exactitude.

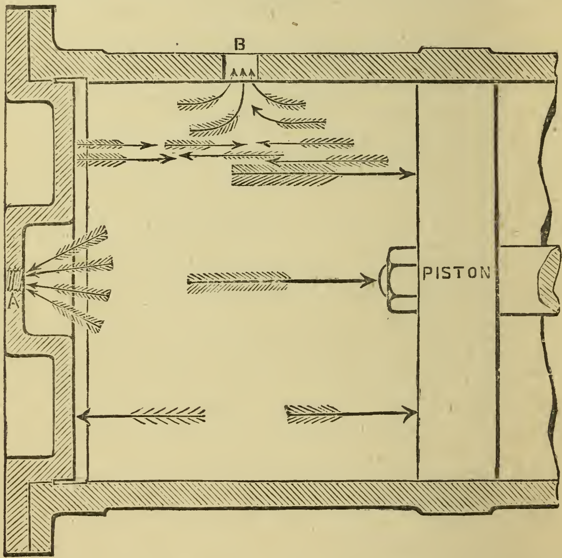
CHAPTER II.

HOW TO TAKE AN INDICATOR DIAGRAM CORRECTLY.

The first matter for notice is the arrangement of the gear, and the position of the instrument. It must be remembered that it is the action of the steam inside the engine cylinder we wish to portray: consequently the motion for the indicating paper or card, *must* be derived from the engine piston rod, as it is the action of the steam on that piston we wish to learn. Starting then with this rule without exception, we must next note the correct position of the indicator on the engine cylinder before alluding to anything else. The indicator should be secured to the *end* of the cylinder, *never* on the *side:* for this reason, the steam that is impelling the piston is also rushing along with it, and when exhaustion ensues, the volume has a retrograde motion, but still in a line with the cylinder's length. The illustration Fig. 3 will assist to understand this matter: the motion and pressure of the steam on the piston and cover

are shown by the large straight arrows. The opening A is *the* situation for the indicator nozzle to be fitted into to take a correct indication of the action of the steam within the cylinder;

Fig. 3.



An Illustration of the action of Steam when propelling the Piston, and showing also the correct position for the Indicator.

the smaller straight arrows, at angles, show that the steam alters its line of motion but *very slightly* to pass through the opening A, because that opening is in a line with the motion of the

steam *in* the cylinder. Notice next the opening B, and the form of the arrows there, which indicate that the steam has actually to *stop* and turn at right angles to pass through that aperture, so that the *actual* pressure or motion of the steam in the cylinder can *never* be known from the *branch* volume issuing at *right* angles to the cylinder's length.

The position of the indicator beyond the nozzle *must* be *close* to it to ensure a truthful indication of the steam in the engine cylinder. If a branch pipe is requisite, let it be as straight as possible, to form a *direct* communication. When the indicator is secured at right angles with the nozzle pipe, of course a bend branch is requisite; let this be *always* a curve whose radius is from four to six inches—never a right angle bend.

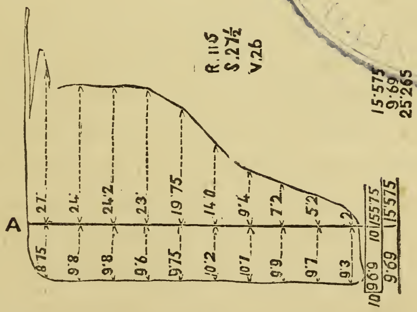
Having supposed the indicator to be fixed, and in connection with the cylinder, our next determination is the best mode to give motion to the paper or card. We have already stated that the original movement must be from the engine piston rod, and there only remains now to explain how to reduce that travel to suit the indicator or card barrel. Now this barrel must not of course make an entire revolution,

for the obvious reason that, if it did, the diagram taken would be connected at its end extremities; the final motion therefore depends on the diameter of the pulley, at the base of the barrel. The general length for this movement is about 4 to 5 inches, the circumference of the pulleys being of corresponding dimensions, plus the space required between the ends of the diagram described. The motion from the engine piston-rod is reduced in length by levers of such proportions as required for that purpose; for example, if the stroke of the engine is 30 inches, and the length of the diagram is to be 5 inches, then the lengths of the levers are as 1 is to 6, or if only one lever is used, then the indicator's motion must be taken from a point on the lever sufficiently below its fixed end to obtain the reduced travel required. The mechanical connection of the lever and the indicator barrel is usually a cord of the kind known as "whipcord," its normal elasticity being expended, by sufficiently stretching it, before its application for the purpose required.

Supposing now that all is ready, the diagram is taken in this manner,—the communication pipe between the engine cylinder and the indicator is blown through by opening the plug-hole,

after which it is closed. Next, the cord is connected to the motion-barrel, the paper of course being in its place: the pointer is made to form the atmospheric line: this motion, without the steam, being permitted for a few times, the plug is again opened, and the diagram described by the pointer as illustrated by Fig. 4. The definition of this diagram is expressed by the author in his work on "Marine Engineering" thus :—

Fig. 4.



Definition of an Indicator Diagram from actual practice.

At the left hand the pointer commenced from the atmospheric line A ; it rose instantly to the highest point, and as quickly fell, and performed the undulation to 27·. It described from thence the uneven line to 23·, when the steam was cut

off. Expansion now commenced, and as a decrease in the pressure of the steam in the cylinder ensued, the pointer gradually fell from from 23· to 14·, when the uneven line was traced to 9·4. Expansion now ceased, and as at the same instant exhaustion commenced, the pointer gradually fell to the "atmospheric line" before the end of the stroke, describing below it the undulated line until the completion. On turning the corner—the return action here commenced—the pointer marked its duty by an uneven line until reaching 9·8. Here compression commenced, and continued until the steam was readmitted into the cylinder termed the "lead," which produced the round corner at the right hand. The pointer next ascended and joined the line it commenced.

The motion in this case was taken from the engine piston-rod, and reduced by a lever, the latter being connected to the indicator by a cord of the usual kind.

The class of engine alluded to is a direct-acting single piston-rod horizontal engine of the usual arrangement for screw propulsion; but as there are other classes of engines both on land and water, we must not omit to describe the mode of taking an indicator diagram from each type.

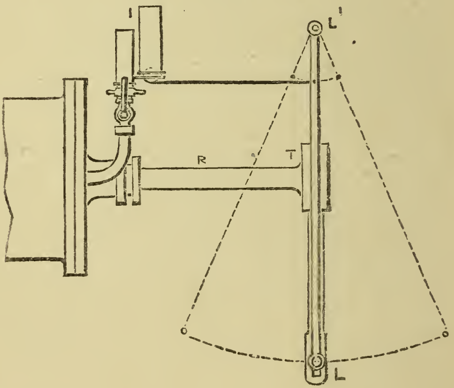
As the directions given about the position of

the indicator are in connection with its movements, we shall now follow on to a few hints as to the most correct method to obtain a true motion for the card-barrel. For steam-engines, whether over-head or side-lever, the cord can be attached either sufficiently near the centre of the gudgeon or to a relatively suitable point on the radius-bar, of course taking precaution in each case to allow no back-lash, slack-string, or non-coincident motion for the barrel with the engine piston-rod. The indicator is secured to the end of the cylinder; and a motion-pulley, situated at a suitable distance, guides the cord from the engine-motion to the pulley of the card-barrel; levers, therefore, are not required.

Now for horizontal, vertical, inclined, or angular, direct, or return-acting engines, the case is widely different; for with these the original motion for the barrel *must* be taken from the piston-rod, because there is no other direct sliding point of any detail coincident with its motion. There are, of course, various methods of arriving at this means; one is as illustrated by Fig. 5. In this example the indicator, I, is secured to the front of the cylinder; the piston-rod, R, is at half-stroke. The first lever, L, is secured to the head, T, of the piston-rod, and therefore moves

always in a vertical position, with a horizontal motion of the same stroke as the engine piston. The length of this lever is proportioned to the position of the motion-pulley of the indicator; it is looped also at the lower end, to permit sufficient vibration for the pin of the second lever, L^1,

Fig. 5.



Direct-acting Lever Indicator Gear.

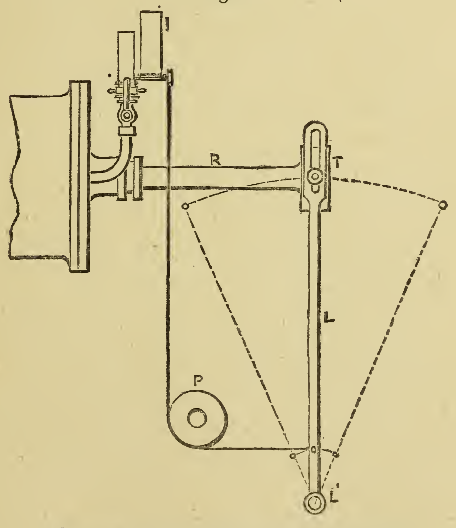
which being hung on a pin secured to a bracket or any other fixture, acts as a pendulum. The final motion, therefore, is derived from a point on the second lever, the chord of the arc described being equal to the length of diagram taken.

A second arrangement for this purpose is illustrated by Fig. 6; in this instance one lever, L,

only is used, and the motion end is looped and attached to the head, T, of the piston-rod, R, being in direct connection with the lever end supported on a fixed pin, L^1.

The indicator, I, is situated as before, but the

Fig. 6.



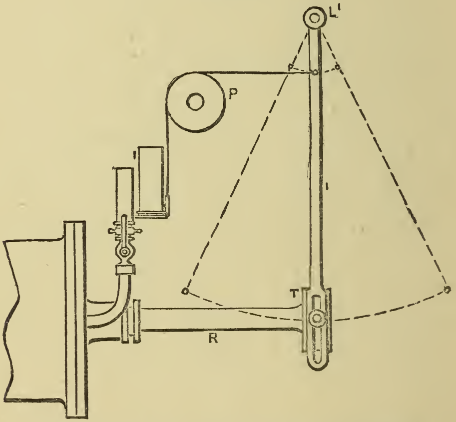
Pulley and Lever Indicator Gear, Under Motion.

motion pulley is at right angles; a second pulley, P, below is introduced to guide the cord to the motion point on the lever, L.

In direct opposition to the last method, as far as the position of the lever is concerned, the

arrangement illustrated by Fig. 7 is often used: here the indicator is in the same position again, but the lever and pulley are reversely situated. The lever, L, is hung on a pin, L^1, supported by a bracket or anything equally applicable: it is looped as before, but attached to the pin on the

Fig. 7.



Pulley and Lever Indicator Gear, Over Motion.

head, T, of the piston-rod, R. The barrel of the indicator, I, derives its motion from the correct point on the lever, L, being connected by a cord passing quarter round the pulley, P, and then continued to the pulley on the barrel.

These three arrangements for the gear are, as already stated, correct for any class of direct or

return-action engine. The main-motion pin for the latter type being fixed on the piston-rod instead of on the head. We may add, that for high speeds the arrangement shown by Fig. 5 in page 18 is the best, because the second pulley, P, in Fig. 6, is dispensed with, its introduction being really a *break* in the direct connection for the cord.

When the performances of oscillating engines require to be indicated, the pulleys *must* be introduced; but as the speed of these engines is never high, comparatively, there is no objection to their use. The best arrangement we know of for this purpose is shown by Fig. 8, which illustrates the half-end elevation of an oscillating cylinder with its piston-rod, and head, T, at full down stroke and the cylinder vertical. The cylinder is fitted with two small steam branch pipes, B B, with bends, *b b*, at each end, connected to the ends; centrally of the length is a two-way cock, C, with a branch, at right angles to which is connected the indicator, I. The motion for the pulley, P, is attained by the cord being attached to the head, T, and led down to the pulley, P^1, which is supported by a bracket secured to the flange of the stuffing-box; following from this, at right angles, the cord is secured at a point on the periphery of the largest pulley, P^2, which has

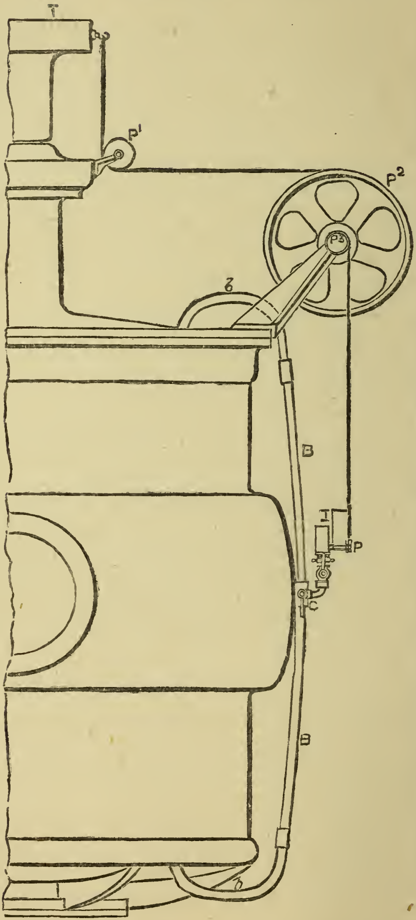


Fig. 8.

Arrangement of Indicator Gear for Oscillating Engines, Vertical position. By Messrs. James Watt and Co.

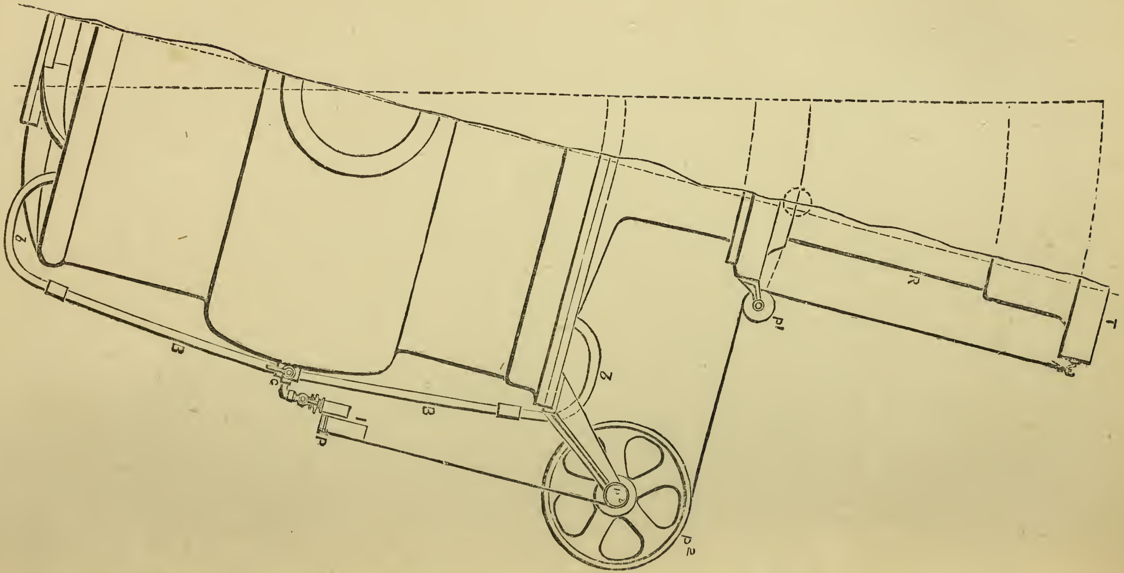
on its shaft another pulley, P³, being supported by a bracket secured to the cylinder. From the

pulley, P^3, a second cord is attached to the barrel-pulley, P, so that a certain connection between P and T is obtained.

Of course the radii of all the pulleys must be in strict ratio; for example, if the motion of the pulley, P, is five inches lineally, and that for the piston-rod head, T, fifty inches, then the radius of the larger pulley, P^2, must be five times more than that of P, P^1, and P^3, supposing each of them to be alike; but if there is any difference in their radii, then the radius of P^2 must be in due proportion. To explain that the gear is independent of any vibratory action of the cylinder, as far as the pulley, P, is affected, the drawing, Fig. 9, is introduced on the next page, which shows the piston-rod, R, at half-stroke, and the cylinder gear and indicator at the relative angle.

The back-lash of the cord is prevented by the pulley, P^2, being fitted with a circular spring, which contracts by the ascent of the head, T, and unwinds when it descends; this pulley is therefore loose on its bearing, and P^3 is fixed to it.

Very little consideration is therefore necessary to understand that, as far as the motion for the indicator is concerned, it must be coincident with the engine piston rigidly so, without slip or faulty return action, the cause of slip being the result of



Arrangement of Indicator Gear for Oscillating Engines, Extreme Angular position. By Messrs. James Watt and Co.

the speed of the piston exceeding that of the indicator, and the faulty return action the jerking of the piston-barrel.

As we shall dilate on the errors and their causes in indicator diagrams further on, we will here point out the list of notes an engineer should make when indicating various engines :—

Notes to be made when indicating Marine Engines.

Name of ship.

Name and locality of trial.

Time and date of ditto.

Type of engines.

Nominal horse-power collectively.

Number of cylinders.

Fore or aft, or port or starboard cylinder indicating.

End of cylinder (top or bottom, back or front).

Effective area of piston in square inches.

Scale of the diagram.

Length of stroke of piston in feet.

Length of crank connecting-rod.

Number of strokes of piston per minute.

Type of slide valve, the dimensions of the steam-ports, and the travel and outside lap of valve, and width of opening for supply of steam.

Class of link motion.

Grade of expansion cut off, if in gear.

Pressure of steam indicated by the gauges in the engine and boiler rooms.

Exhaustion indicated by the vacuum gauge.

Amount of injection-water passage opened.

Feed water, off or on.

Temperature of the engine and stoking-rooms.

State of fires in the boilers.

Height of water in ditto.

Priming, if any symptoms.

State of the working bearings of the engine, and shafting from the forward main bearing to the stern tube stuffing-box.

General appearance of the duty of the engine.

Type of screw-propeller or paddle-wheels, and proportions.

Speed of the ship in knots per hour; her dimensions, draught, tonnage, and displacement.

State of the water and tide.

For Land, Stationary, and Portable Engines.

For land-engines, the portion of the above notes that refer to them must be used, with this addition, for locomotive engines.

The diameter of the driving-wheels.

Weights of the engine and train.

The incline of the line of rails, if any.

The radii of the curves of ditto.

In the case of combined, or high and low pressure steam-engines, the length of the stroke of each piston must be recorded, together with the pressures of the steam in each, if gauges are attached.

There are, of course, with all kinds of engines, other incidents that are exceptional, which the engineer who understands his profession will not fail to notice, but, as they are casual, need no comment here.

CHAPTER III.

THE PROOF OF ATMOSPHERIC PRESSURE, AND PARTICULARS OF STEAM PRESSURES.

The two preceding chapters may be considered as preliminary to the present one; for the mere fact of understanding the mechanical arrangement of an indicator, and the mode of using that instrument properly, do not constitute the entire requirements bearing on the subject. What is exceptional is a correct knowledge of the principles on which the indicator diagram is founded, for by an acquaintance with them the real utility of the instrument becomes apparent.

To begin, then, we must first notice the nature and real cause and effect of atmospheric pressure, the exhaustion of which, in relation to the condensing-engine is termed "vacuum." The utility of this natural phenomenon was experimented on as far back as 1643, by Torricelli, a pupil of the renowned astronomer and philosopher Galileo, in the following manner:—A glass tube, about a yard in length, and of a quarter-inch diameter inside,

was sealed at one end, and quite filled with mercury. The open end being stopped either by the thumb or other efficient means, ready to be removed at pleasure, the tube was then inverted, and the lower end inserted in a trough of mercury; the tube still remaining vertical, and the thumb being removed, or the immersed end opened, the mercury sank six inches from the sealed end, which showed thirty inches of mercury in the tube from the level of that in the trough. The mass in the tube was therefore held by the atmosphere, or the pressure of the latter on the mercury in the trough prevented that in the tube descending below the six inches alluded to. Now if the top end of the tube had been opened, the mercury in it would have sunk into the trough; because the pressure of the atmosphere would have been in equilibrium on the inside and outside of the tube, and thus the mercury would fall, owing to gravity.

The application of this experiment for practical purposes was thus carried out. A tube, *one* square inch in area, filled as before, and inverted likewise in a trough of suitable dimensions, retained the mercury thirty inches high, under the same circumstances. Then, as 2 cubic inches of mercury equals about 1 lb. avoirdupois

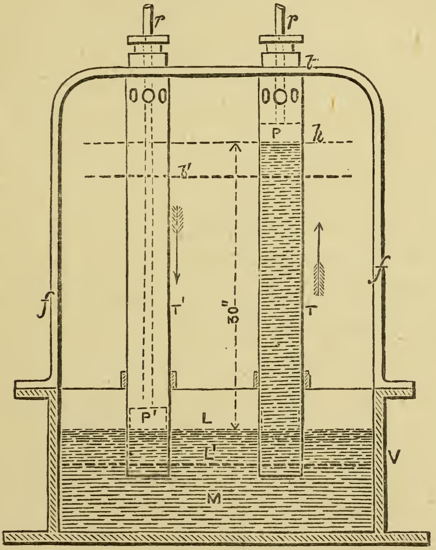
weight, and there are 30 cubic inches resisting the atmosphere, the matter is resolved into a simple calculation, thus :—W = weight of the mercury, in the tube, C = cubic inches of mercury to equal 1 lb., and x = full pressure of the atmosphere in pounds avoirdupois per square inch; therefore, $x = \frac{W}{C}$ or, $\frac{30}{2 = 1 \text{ lb.}} =$ 15 lbs. on the square inch, which is termed a *perfect* vacuum; but 14·9 lbs. is the utmost yet attained actually.

From the above conclusion emanate the well-known symbols, V 26, or V 28, &c., which, in reality, mean that, relatively, a vacuum was obtained equal to 13 or 14 lbs. on the square inch, being the pressure relieved or absorbed by the air-pump in practice.

Another mode of learning the actual pressure of the atmosphere has been often tried, as illustrated by Fig. 10. The vessel V was filled with mercury, M, the vessel being surmounted by a frame $f\,f$, which supported two glass tubes T and T′, perforated at the top. These tubes are fitted with pistons P P′, and rods r r, for the purpose of demonstrating the utility of the apparatus. On raising the piston P to the height h in the tube T, the mercury immediately ascended after it for 30 inches, as shown; the level of the mercury in

the vessel sank to L; while the pressure of the atmosphere not only caused this, but also held or forced the piston P′ down to that level of the mercury in the vessel. Now when P′, in the

Fig. 10.



Practical Method for Testing the Atmospheric Pressure.

tube T′, was raised to *b*′, the level of the mercury in the vessel fell to L′ or V; for as the piston ascended it caused a vacuum behind it, and the mercury in the vessel was forced by the atmospheric pressure up into the tube till reaching 30 inches from the

level L′; while the mercury in the tube T of course sank to the same level *b*′, as that in the tube T′. We again learn from this, as in the preceding experiments, that there can be no other agency for causing the mercury to rise in the tubes than the natural pressure of the atmosphere; for as the pistons were raised, they drove the air in the tube above them out at the apertures at the top, and as there was no air in the tube below, the pressure of the atmosphere acted only on the *outside* of the tube and on the mercury, which resulted in the latter being forced up as shown.

We have now sufficiently explained how it is certain that there is an atmospheric pressure, which in round numbers is about 14·5 lbs. on the square inch, or 29 inches of mercury in general practice, when a nearly perfect vacuum is attained; for undoubtedly the state of the atmosphere in an engine-room is not clear; consequently the natural pressure is proportionately reduced, in accordance.

Our next step in instruction is the explanation of the nature of steam; for, as we stated before, if the young engineer or student wishes to be practically acquainted with any subject properly, he *must* know the principles *first*.

It matters not for the present purpose about

the chemical properties of steam, but rather its "nature" and proportions in particular, which are given by the following table :—

Table of Saturated Steam at different Pressures.

Actual pressure of the Steam exclusive of the Atmospheric Pressure.	Temperature in degrees of Faht.	Elastic force in inches of Mercury.	Cubic inches of Water to produce 1 cubic foot of Steam.	Volume of Steam produced from one of Water.	Weight of one cubic foot of Steam in lbs.
lbs.					
1	216·3	32·64	1·099	1572	·0397
2	219·5	34·68	1·162	1487	·0419
3	222·5	36·72	1·226	1410	·0442
4	225·4	38·76	1·287	1342	·0465
5	228·0	40·80	1·350	1280	·0487
6	230·6	42·84	1·411	1224	·0510
7	233·1	44·88	1·474	1172	·0532
8	235·5	46·92	1·536	1125	·0576
9	237·9	48·96	1·597	1082	·0598
10	240·2	51·00	1·658	1042	·0620
11	242·3	53·04	1·719	1005	·0642
12	244·4	55·08	1·779	971	·0664
13	246·4	57·12	1·840	939	·0686
14	248·4	57·16	1·910	909	·0707
15	250·4	61·20	1·959	881	·0729
16	252·2	63·24	2·021	885	·0751
17	254·1	65·28	2·079	830	·0772
18	255·9	67·32	2·138	807	·0794
19	257·6	69·36	2·198	785	·0815
20	259·3	71·40	2·258	765	·0837
21	260·9	73·44	2·316	745	·0858
22	262·6	75·48	2·376	727	·0879
23	264·2	77·52	2·433	709	·0900
24	265·8	79·56	2·493	693	·0921

D

Actual pressure of the Steam exclusive of the Atmospheric Pressure.	Temperature in degrees of Faht.	Elastic force in inches of Mercury.	Cubic inches of Water to produce 1 cubic foot of Steam.	Volume of Steam produced from one of Water.	Weight of one cubic foot of Steam in lbs.
lbs.					
25	267·3	81·60	2·552	677	·0942
26	268·7	83·64	2·610	661	·0963
27	270·2	85·68	2·670	647	·0983
28	271·6	87·72	2·725	634	·1004
29	273·0	89·76	2·787	621	·1025
30	274·4	91·80	2·842	608	·1046
31	275·8	93·84	2·899	595	·1067
32	277·1	95·88	2·958	584	·1087
33	278·4	97·92	3·015	573	·1108
34	279·7	99·96	3·074	562	·1129
35	281·0	102·00	3·130	552	·1141
36	282·3	104·04	3 188	542	·1150
37	283·6	106·08	3·248	532	·1170
38	284·7	108·12	3·304	523	·1192
39	285·9	110·10	3·361	514	·1212
40	287·1	112·20	3·415	506	·1232
41	288·2	114·24	3·465	498	·1252
42	289·3	116·28	3·526	490	·1272
43	290·4	118·32	3 585	482	·1292
44	291·6	120·36	3·645	474	·1314
45	292·7	122·40	3·700	467	·1336
46	293·8	124·44	3·756	460	·1356
47	294·8	126·48	3·814	453	·1376
48	295·9	128·52	3·865	447	·1396
49	296·9	130·56	3·927	440	·1416
50	298·0	132·60	3·981	434	·1436
51	299·0	134·64	4·037	428	·1456
52	300·0	136·68	4·094	422	·1477
53	300·9	138·72	4·143	407	·1497
54	301·9	140·76	4·204	411	·1516
55	302·9	142·80	4·256	406	·1535
56	303·9	144·84	4·309	401	·1555

Actual pressure of the Steam exclusive of the Atmospheric Pressure.	Temperature in degrees of Faht.	Elastic force in inches of Mercury.	Cubic inches of Water to produce 1 cubic foot of Steam.	Volume of Steam produced from one of Water.	Weight of one cubic foot of Steam in lbs.
lbs.					
57	304·8	146·88	4·363	396	·1574
58	305·7	148·92	4·419	391	·1595
59	306·6	150·96	4·476	386	·1616
60	307·5	153·00	4·535	381	·1636
61	308·4	155·04	4·583	377	·1656
62	309·3	157·08	4·645	372	·1675
63	310·2	159·12	4·695	368	·1696
64	311·1	161·16	4·747	364	·1716
65	312·0	163·29	4·812	359	·1736
66	312·8	165·24	4·867	355	·1756
67	313·6	167·28	4·923	351	·1776
68	314·5	169·32	4·965	348	·1795
69	315·3	171·36	5·023	344	·1814
70	316·1	173·40	5·082	340	·1833
71	316·9	175·47	5·127	337	·1852
72	317·8	177·48	5·189	333	·1871
73	318·6	179·52	5·236	330	·1891
74	319·4	181·56	5·300	326	·1910
75	320·2	183·60	5·349	323	·1929
76	321·0	185·64	5·400	320	·1950
77	321·7	187·68	5·451	317	·1970
78	322·5	189·72	5·520	313	·1990
79	323·3	191·76	5·574	310	·2010
80	324·1	193·80	5·628	307	·2030
81	324·8	195·84	5·665	305	·2050
82	325·6	197·88	5·721	302	·2070
83	326·3	199·92	5·779	299	·2089
84	327·1	201·96	5·837	296	·2108
85	327·8	204·00	5·897	293	·2127
90	331·3	214·20	6·150	281	·2218
95	334·6	224·40	6·424	269	·2317
100	338·0	234·60	6·672	259	·2406

The use of the above table for practical purposes is this:—Suppose the capacity of a cylinder is 80 cubic feet, and the steam is cut off from it when the piston has made $\frac{1}{4}$th of the complete stroke, then $\frac{80}{4} = 20$ cubic feet of space, which is the amount of steam contained in the cylinder during one stroke of the piston. We will imagine that the *full* pressure of the steam is 60 lbs. on the square inch, and next, that we require to know the amount of water which the steam in question represents; looking at the table, we see that one cubic foot of steam at 60 lbs. pressure is produced from 4·535 cubic inches of water, then $20 \times 4{\cdot}535 = 90{\cdot}7$ cubic inches of water, in the form of steam, used *in the cylinder* for each stroke of the piston.

Next we must notice the elasticity of the steam, and its temperature corresponding with the pressure. Conclude, as an example, that the steam is 40 lbs. on the square inch, we shall see in the table that its temperature is 287·1°. Suppose the 40 lbs. to be the pressure in the boiler, and that it loses 5 lbs. on the square inch when reaching the cylinder, what will the loss of heat be? Here let $40 - 5 = 35$; then 35 lbs. pressure, according to the table, $= 281°$, which shows

that 287·1° − 281 = 6·1° loss of heat, or what is lost by radiation.

Imagine next that the steam, at 35 lbs. pressure, entering the cylinder, is cut off at $\frac{1}{4}$th of the stroke of the piston, and is exhausted at $\frac{3}{4}$ths of the stroke, what will be the reduction of the pressure at the point of exhaustion? The area of the cylinder is assumed as 20 feet, the stroke of the piston 4 feet; then $20 \times 1 = 20$ cubic feet, being the amount of steam in the cylinder when the supply terminated; the exhaustion occurring when the piston has reached 3 feet. Now, as the theoretical law which governs the rate of the expansion of steam is, that inversely as the cubical contents of the space in the cylinder for expansion increases, so will the pressure of the steam in it be reduced, supposing the supply to be cut off. This is founded on the law discovered by Boyle and Mariotte, who proved "that the volume of a given quantity of steam is inversely as the pressure it bears;" for example, if steam, at 60 lbs. on the square inch, occupying 20 cubic feet, is expanded into 40 cubic feet, the pressure will be reduced to one-half, or 30 lbs.; which converts the following formula introduced by the author in his paper read at the Hall of the London

Society of Arts, to the members, December 18th, 1867:—

"To the cubical contents for the supply, add the cubical contents for expansion; divide the total sum by the cubical contents for supply; the initial pressure of the steam, divided by the quotient, equals the pressure of the steam when expansion terminates. Now, let A = cubical contents for supply; B = cubical contents for expansion; C = initial pressure of the steam; and x = the pressure when expansion terminates; then, by the above formula, the symbols are thus arranged, $x = C \div \left[\frac{A+B}{A}\right]$; which, if put into the figures alluded to in the question, are thus—

$$\left.\begin{array}{l} A = 20 \\ B = 40 \\ C = 35 \end{array}\right\} 20 + 40, \text{ then } \frac{60}{20} = 3, \text{ and } \frac{35}{3} = 11{\cdot}666$$

for the pressure of the steam when expansion is concluded."

It will, therefore, be understood that A = the area of the cylinder, × the length of the stroke of the piston during the supply of the steam, and that B = the length from the end of A, that the piston travels till exhaustion ensues, × by the same area, and that C is the maximum pressure of the steam in the cylinder.

Now, on referring again to the table, we shall find the relation that the volumes of the steam bear to each other under the two pressures named; at 35 lbs. the volume is 552, and at 11 lbs. the volume is 1005, and at 12 lbs. it is 971, so that the mean $= \frac{1005 + 971}{2}$, 988; therefore, 11·666 lbs. pressure corresponds with a volume of about 996; then $\frac{996}{552} = 1{\cdot}80$, which proves that the steam is reduced in rarefication in the proportion of as 1 is to 1·80, or that the saturation of the steam is 1·80 times more apparent at 11·666 lbs. pressure than at 35 lbs. on the square inch, while the temperatures are 243·4° and 281° respectively, showing a reduction of 37·6° of heat, radiation excepted.

Leaving the table for the present, we now direct attention to the mean pressure of the steam in the cylinder from the point of admission and exhaustion. The mode of arriving at this conclusion approximately is to divide the length of the stroke of the piston during expansion into a certain number of parts, and by the preceding formula produce the sums of the pressures at those points; add the whole together, and by dividing the result by the numbers of the divisions, the mean pressure can be known; but

this mode is rather approximate for the purpose of learning the actual mean pressure. Another method is to introduce hyperbolic logarithms into the formula, because the curve of expansion is said to be hyperbolic, which we have explained in the next chapter, and also shown the geometrical means of producing it.

Hyperbolical Rule to produce the Sum of the Mean Force or Pressure of the Steam in lbs. per Square Inch for one Stroke of the Piston.—

Divide the length of the stroke in inches by the distance in inches the piston moves from the commencement of its stroke to the point of cutting off the steam, and divide the maximum pressure by the quotient; then add 1 to the hyperbolic logarithm of the *first* quotient, which is the number of times the steam has expanded, and multiply the sum by the *second* quotient, which is the number of lbs. to which the steam is expanded, and the product is the final result. Example:—

Maximum pressure of steam . .	50 lbs.
Length of stroke of piston. . .	40 inches.
Length of stroke to cut off . .	10 inches.

Then 40 in. $\div$ 10 = 4, the first quotient, and 50 lbs. $\div$ 4 = 12·5, the second quotient; then

hyp.-log. of 4 = 1·386 + 1 = 2·386, and 2·386 × 12·5 = 29·825 lbs. mean pressure.

Table of Hyperbolic Logarithms.

No.	Hyperbolic Logarithm.	No.	Hyperbolic Logarithm.	No.	Hyperbolic Logarithm.	No.	Hyperbolic Logarithim.
1	·0000	$3\frac{3}{4}$	1·32175	$6\frac{1}{2}$	1·87180	11	2·39790
$1\frac{1}{4}$	·22314	4	1·38629	$6\frac{3}{4}$	1·90954	12	2·48491
$1\frac{1}{2}$	·40546	$4\frac{1}{4}$	1·44691	7	1·94591	13	2·56495
$1\frac{3}{4}$	·55961	$4\frac{1}{2}$	1·50507	$7\frac{1}{4}$	1·98100	14	2·63906
2	·69315	$4\frac{3}{4}$	1·55814	$7\frac{1}{2}$	2·01490	15	2·70805
$2\frac{1}{4}$	·81093	5	1·60944	$7\frac{3}{4}$	2·04769	16	2·77259
$2\frac{1}{2}$	·91629	$5\frac{1}{4}$	1·65822	8	2·07944	17	2·83321
$2\frac{3}{4}$	1·01160	$5\frac{1}{2}$	1·70474	$8\frac{1}{2}$	2·14006	18	2·89037
3	1·09861	$5\frac{3}{4}$	1·74919	9	2·19722	19	2·94444
$3\frac{1}{4}$	1·17865	6	1·79176	$9\frac{1}{2}$	2·25129	20	2·99573
$3\frac{1}{2}$	1·25276	$6\frac{1}{4}$	1·83258	10	2·30259	21	3·04452

There is, of course, no allowance made for the loss of heat or reduction of the pressure in either of the preceding or the following rules.

PROFESSOR RANKINE'S RULES FOR THE MEAN PRESSURE.—These formulæ are to obtain the ratio in which the *mean* absolute pressure will be less than the *initial* absolute pressure, being nearly exact for dry saturated steam.

(1). $$\frac{P\,m}{P\,1} = \frac{17 - 16\,r - \frac{1}{16}}{r}$$

For moderately moist steam—

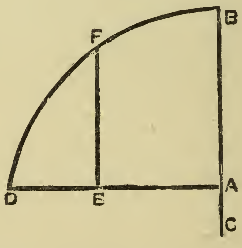
$$(2).\qquad \frac{P\ m}{P\ 1} = \frac{1 + \text{hyp.-log. } r}{r}$$

Where r = rate of expansion; where – P 1 = initial absolute pressure, and P m = mean absolute pressure.*

The Professor has also a geometrical method, as shown by Fig. 11, and described as follows.—

Draw a straight line C A B, in which make A B = 4 A C. Draw A D perpendicular to C A B; and about C describe the circular arc B D cutting A D in D.

Fig. 11.



Professor Rankine's Diagram of the Expansion of Steam.

Then in D A take E, so that $\frac{D\ E}{D\ A}$ shall represent the *effective cut-off* (and consequently $\frac{D\ A}{D\ E}$ the rate

* The quantity $r - \frac{1}{16}$ is known by taking the reciprocal of r, and extracting the square root of it four times

of expansion). At E draw E F parallel to A B. Then $\frac{E\,F}{A\,B}$ will be the required ratio of mean to initial absolute pressure, nearly.

SIMPSON'S RULE FOR THE MEAN PRESSURE.—To the sum of the extreme pressure per square inch, add four times the sum of the even pressures, and twice the sum of the odd pressures; then this sum, multiplied by one-third of the distance between the consecutive points at which the pressures are taken, will give the work done expansively per square inch of the area of the piston in one stroke.

ORDINARY RULE FOR THE MEAN PRESSURE.—To the pressure of the steam at the point of cut-off add the mean of the divisional pressures during expansion; the sum divided by the numbers of the different pressures equals the mean pressure of the steam.

Example,—

Maximum pressure of steam . . .	60 lbs.
Length of stroke of piston during supply	12 in.
Length of stroke of piston during expansion	12 in.

	Supply.	Expansion.				
Motion of piston	12 in.	3	3	3	3	Exhaustion.
Ratios of position	0	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	
Initial pressures of steam	60	48	40	34·28	30	
Mean of divisional pressures	54	44	37·14	32·14		
Mean of initial pressures		42·45				

The meaning of this table is this :—when the piston travelled 12 inches, the steam was 60 lbs. on the square inch ; on its moving 3 inches more, which was $1\frac{1}{4}$ of the amount of the stroke for supply, the pressure was reduced to 48 lbs. from the formula $\frac{60}{1·25} = 48$. The piston then moved 3 inches more, this position being $1\frac{1}{2}$ of the length for supply, and again the formula $\frac{60}{1.5} = 40$; another 3 inches advance, and $1\frac{3}{4}$ of the supply stroke was made, and the pressure was then known from $\frac{60}{1·75} = 34·28$; and, lastly, the 12 inches were completed, and the ratio was twice ; then $\frac{60}{2} = 30$ being the pressure of the steam at the point of exhaustion. Next comes the application of the rule, $60+48+40+34·28+30=$ $212·28$, then $212·28 \div 5 = 42·45$, which proves

that the mean of initial pressures at the five points from the beginning of the stroke to the point of exhaustion equals 42·45 lbs. The mean divisional pressures are given more for comparison than use in the formulæ.

CHAPTER IV.

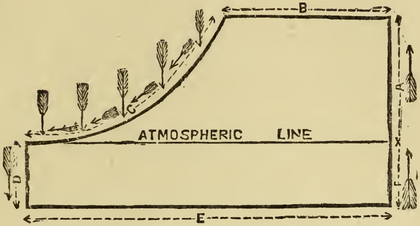
THE THEORETICAL GEOMETRY OF THE INDICATOR DIAGRAM.

It will be remembered that in Chapter I. we explained the arrangement of the indicator and its use. Chapter II. we devoted to the mechanical questions of the gear, and in the preceding chapter to this we dwelt on the matters of atmospheric and steam pressures in connection with the steam-engine; we have not therefore yet touched on the theory of the form of the diagram, which subject is in its proper place as this chapter.

It has been proved for some years past that the true curve the indicator pencil should trace on the card, when the steam is expanding in the cylinder, is hyperbolical; and as the remainder of the pencilled figure is a portion of a parallelogram, the curve is the only geometrical question of any note in it. As a proof of this, we introduce Fig. 12.

The explanation of this is, that when the pencil was stationary, the atmospheric line was described straight, because there was no steam

Fig. 12.



A practical Definition of the Theoretical Form of an Indicator Diagram.

pressure to start the indicator piston; but when the steam acted on it, the pencil rose vertically for the length or height of A. At this point, it must be remembered, the motion for the indicator barrel commenced, and therefore, as the pencil was held still by the steam acting on the piston during the supply, again a straight horizontal line, for the length of B, was traced; here the steam was cut off from the cylinder, and the expansion commenced, which of course gradually reduced the pressure; and as the card-barrel was still turning, the pencil described a curve as it descended, until reaching the atmospheric line. Here the card-barrel stopped, and the pencil continued to fall continuously at right angles to the steam line B, until the vertical line D was traced, which indicated the amount of vacuum attained in the cylinder; the pencil then became stationary, for the atmospheric pressure forced the indicating

piston down and kept it in that position while the vacuum lasted, as firmly as the steam held it up while the steam pressure remained, so that the pencil is always still when the *full* pressures are on, either under or over the piston. When the pencil stopped at the lower end of the line D the barrel began to turn, but in the opposite direction, and thus the line E was traced, at the end of which the barrel again stopped, and when the steam entered the cylinder the pencil instantly rose, and completed the diagram by forming the line F.

The motions for the pencil are therefore thus arranged: A, F, and D are instantaneous vertical movements; C is a gradual descent; while B and E have *no* motion.

It is also evident that A is admission, B supply, C expansion, and often a portion of it—before joining the atmospheric line—is exhaustion also. D sudden exhaustion, E continuous steady exhaustion, and F re-admission. Then, by knowing this, we can learn also that, while there is admission A, the pencil moves, as indicated by the arrow, and stops at the top end; the supply B is fixed, or a stationary position for the marker, therefore there is no arrow; C, the expansion, is shown by the arrows as being pro-

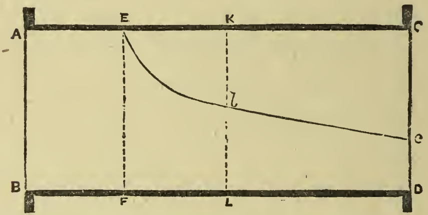
duced by the pencil gradually descending: D, the sudden exhaustion, is where the pencil stopped at the end of C, and fell at once in the direction of the arrow, to the line E; it then became stationary, therefore the arrow is omitted again; after this E, continuous exhaustion, was traced; when F, re-admission, shows by the arrow that the pencil rose vertically, as it did at A. This proves also that the lines B and E are portions of a circle; D, F, and A, vertical straight lines; and C, a hyperbolical curve.

The theory of the indicator diagram is thus proved to be the simple fact that *pressure under the indicating piston causes the pencil to rise*; whether that pressure is steam, water, or air, it matters not, as the *force* is the main agent; and that immediately the *decrease of pressure occurs, the pencil descends.* So that if the young engineer will note this in his memory, he will save himself, and others he is professionally connected with, a vast amount of trouble.

Our next step is the explanation of the hyperbolical curve of expansion, which Watt agreed to be correct at the time he invented the indicator; his theory is described by Mr. J. Scott Russell, M.A., Engineer, in his work on the Steam Engine, as far back as 1841.

"Let ABCD (Fig. 13) represent a section of the cylinder of a steam-engine, and EF the surface of its piston. Let us suppose that the steam was admitted while EF was in contact with AB,

Fig. 13.



Watt's Method of Illustrating the Expansion of Steam.

and that as soon as it had pressed it down to the situation EF, the steam-cock is shut. The steam will continue to press it down, and, as the steam expands, its pressure diminishes. We may express its pressure (exerted all the while the piston moves from the situation AB to the situation EF) by the line EF. If we suppose the elasticity of the steam proportional to its density, as is nearly the case with air, we may express the pressure on the piston in any other position such as KL or DC, by L l and D c, the ordinates of a rectangular hyperbola E l c, of which AE AB are the asymptotes, and A the centre. The

accumulated pressure during the motion of the piston from E F to DC will be expressed by the area EFcDF, and the pressure during the whole motion by the area ABFcDA.

"Now it is well known that the area EFcDF is equal to ABFE multiplied by the hyperbolic logarithm of $\frac{AD}{AE} = L\frac{AD}{AE}$, and the whole area ABE$c$DB is $=$ ABFE $\times \left(1 + L\frac{AD}{AE}\right)$.

"Thus let the diameter of the piston be 24 inches, and the pressure of the atmosphere on a square inch be 14 lbs.; the pressure on the piston is 6333 lbs. Let the whole stroke be 6 feet, and let the steam be stopped when the piston has descended 18 inches, or 1·5 feet. The hyperbolic logarithm of $\frac{6}{1 \cdot 5}$ is 1·3862943. Therefore the accumulated pressure ABEcDB is $=$ $6333 \times 2 \cdot 3862943 = 15112$ lbs.

"As few professional engineers are possessed of a table of hyperbolic logarithms, while tables of common logarithms are, or should be, in the hands of every person who is much engaged in mechanical calculations, let the following method be practised :—Take the common logarithm

of $\frac{BD}{AE}$ and multiply it by 2·3026, the product is the hyperbolic logarithm of $\frac{AD}{AE}$.

"The accumulated pressure while the piston moves from AB to EF is 6333×1, or simply 6333 lbs. Therefore the steam while it expands into the whole cylinder adds a pressure of 8781 lbs.

"Suppose that the steam had got free admission during the whole descent of the piston, the accumulated pressure would have been 6333×4, or 25332 lbs."

Now accepting this hypothesis as correct, which it is in the main, but subject of course to various deviations of importance, to which we shall refer as we proceed, we next direct attention to a practical demonstration of the relation that the indicator diagram bears to the engine cylinder.

The present general practice to define this has been to make a section of the cylinder, and draw an outline of the diagram within its limits. Now nothing possible can be more faulty than this method, because the diagram in practice is never formed to scale, so that its height equals the diameter of the cylinder, and its length equal

to the stroke of the piston. It becomes then important to remove this erratical mode, and introduce a correct one. But before we illustrate this we will dwell a little on the matter, as a discussion.

First. What is the meaning of the height of the indicator diagram? It is the extent that the pencil moves up or down according to the pressure on the piston, which movement above the atmospheric line depicts steam pressure, and below it natural pressure or atmospheric.

Second. What is the definition of its length? Of course a direct answer to this is—the limit of the motion given by the card-barrel; but this is not all; it involves also that, inasmuch as the movement of the barrel is proportionate to the stroke of the piston, the length of the diagram represents that stroke. Then if the length of the movement of the barrel has nothing to do with the pressure in the cylinder, which it has not, to suppose that the height and length of the diagram, *taken at the same scale,* represents the cylinder in its correct proportion is fallacious in the extreme.

Third. Then what is the correct way of representing the connection of these matters? It is to know that, as the *horizontal* length of the

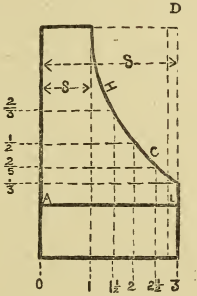
diagram above the atmospheric line indicates the greatest *lineal* equal pressure in the cylinder, it must bear a definite relation to the pressure of the steam only, and *not* therefore to the diameter of the cylinder at all; simply because the pencil is held by the pressure, and that the force would be the same in a large as in a small cylinder, if it were supplied from the boiler proportionally, which of course it always is; hence this axiom—the indicator diagram, in practice, represents the action of the steam in the cylinder; its properties are that the height above the atmospheric line represents the greatest pressure, *without relation to the diameter of the cylinder*; but its length is a direct proportion to the stroke of the piston, and that the scale for the height bears no reference to that for the length.

If, however, it is preferred to arrange the theoretical diagram within the cylinder proportionately to it, the method is as illustrated by Fig. 14.

Here S equals the stroke of the piston, the cubical contents of the steam passage, and piston clearance, and D the diameter of the cylinder, at a scale of $\frac{1}{4}$ inch = 1 foot; *s* is the length of supply-steam line, the steam being cut off at one-third of the stroke; HC is the hyperbolical curve of expansion, and AL the

atmospheric line. The production of HC is of course the only purpose in question, and it is thus obtained :—The spaces 0 to 1, 1 to 2, and 2 to 3 are each equal to *s*. The total pressure of D is 60 lbs. on the square inch; the

Fig. 14.



Theoretical Indicator Diagram within a Cylinder, the Diameter = D, and the Length = S.

height of the first ordinate is equal to D; the second equals $\frac{60}{1\cdot5}$ = 40 in., because it is $1\frac{1}{2}$ of 0 to 1 from 0; then, as 2 is 2 of 1 to 0 from 0, its height is $\frac{60}{2}$ = 30 in.; next, $2\frac{1}{2}$ equals $\frac{60}{2\cdot5}$ = 24 in., and 3 is 20 in. drawn from $\frac{60}{3} = 20$, which, of course, is the lowest pressure; then, where the horizontal lines $\frac{2}{3}$, $\frac{1}{2}$, $\frac{2}{5}$, and $\frac{1}{3}$ intersect with the ordinates $1\frac{1}{2}$, 2, $2\frac{1}{2}$, and 3, are the

points through which the line HC is drawn to join the top of the ordinate 1.

It is obvious that the principle of this method is founded on the law of Mariotte, which is, "that the volume of a given quantity of steam is inversely as the pressure it bears," as we expressed in page 37. Similarly, therefore, in all cases, if D is divided by the relative position of any ordinate to the length of *s*, the quotient equals the height of that ordinate; then the height of $[1 = D]$, $[1\frac{1}{2} = D \times 2 \div 3]$, $[2 = D \div 2]$, $[2\frac{1}{2} = D \times 2 \div 5]$, and $[3 = D \div 3]$; therefore, irrespective of *any* pressure being taken into consideration, the line HC can be drawn correctly. But suppose the pressure is considered, it will be expressed thus :—

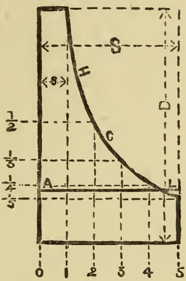
				Ratios.			
Positions of the piston in relation to its stroke. . .	1	$\frac{1}{3}$	$\frac{1}{3\cdot5}$	$\frac{2}{3}$	$\frac{2}{3\cdot5}$	$\frac{3}{3}$,	or full stroke.
				Pressures.			
Pressure of the steam in lbs. per square inch at these points . .	60	60	40	30	24	20,	exhaustion.

Or it may be expressed in this manner,—

			Pressures.			
Pressure of steam . . .	60	60	40	30	24	20 lbs.
			Ratios.			
Lengths of the ordinates in proportion to the diameter (D) of the piston.	1	1	$\frac{2}{3}$	$\frac{1}{2}$	$\frac{2}{5}$	$\frac{1}{3}$.

As a matter of comparison, we next introduce Fig. 15. This illustrates that the proportions

Fig. 15.



Theoretical Indicator Diagram within a Cylinder, the Diameter = D, and the Length = S.

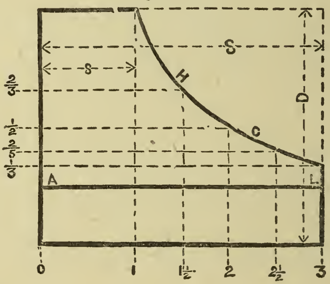
of D and S are as before; but the length of *s* is much less, or $\frac{1}{5}$ of S; then, as *s* in Fig. 14 was $\frac{1}{3}$ of S, and the main ordinates 1, 2, and 3 were repetitions of position, so are the ordinates 1, 2, 3, 4, and 5 equidistant as 0 to 1 in Fig. 15. Here, also, as the positions of the ordinates are whole numbers, so their heights are even fractions of D, in this manner :—

			Numbers.			
Positions of the ordinates or piston, as numbered	0	1	2	3	4	5, end.
			Ratios.			
Proportions of their positions to S, or stroke of the piston	0	1	$\frac{1}{2}$	$\frac{1}{3}$	$\frac{1}{4}$	$\frac{1}{5}$, nil.
			Pressures.			
Pressures of the steam in lbs.	60	60	30	20	15	12, exhaustion.

This defines that the exhaustion of the steam ensues at a pressure of only 12 lbs. on the square inch, by cutting off at $\frac{1}{5}$ of S the stroke of the piston; whereas in Fig. 14 the lowest pressure was nearly double at the same point, or 20 lbs.

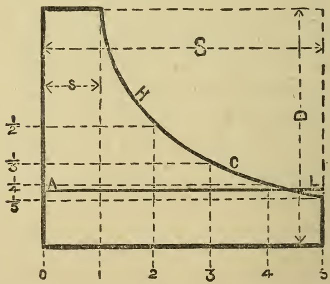
To further elucidate this matter, we illustrate another diagram by Fig. 16. Here the diameter

Fig. 16.



Theoretical Indicator Diagram within a Cylinder, the Length = S, and the Diameter = D.

Fig. 17.

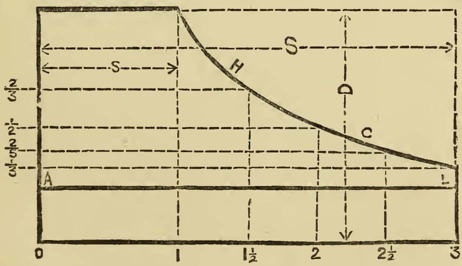


Theoretical Indicator Diagram within a Cylinder, Length = S, and Diameter = D.

D is 60, but the stroke S is 72 instead of 36, as in Figs. 14 and 15. The length of s being again $\frac{1}{3}$ of S, all the ordinates in length and position are precisely as before, with the same ratios and pressures. As a comparison, Fig. 17 is introduced, where s is $\frac{1}{5}$ of S, and D 60, while S is 72, their ordinates and their ratios being figured as in Fig 15. The table in page 57 applies in this case also.

There not being much difference in the lengths, and none with the diameter D in the two last examples, we next illustrate one with a greater contrast by Fig. 18.

Fig. 18.

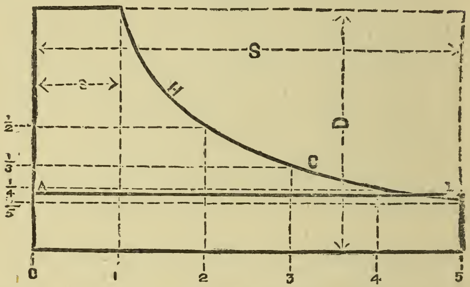


Theoretical Indicator Diagram within a Cylinder, the Length = S, and the Diameter = D.

Here D is 60, and S is 108, or three times of s, as in Fig. 14; but as the cut-off, however, is of the same ratio, the ordinates and pressures are similar also.

Next, Fig. 19 depicts a contrast with Fig. 15; here, too, the length of *s* is the same in proportion to S as in the second example, and all the other matters here are as there.

Fig. 19.



Theoretical Indicator Diagram within a Cylinder, the Length = S, and the Diameter = D.

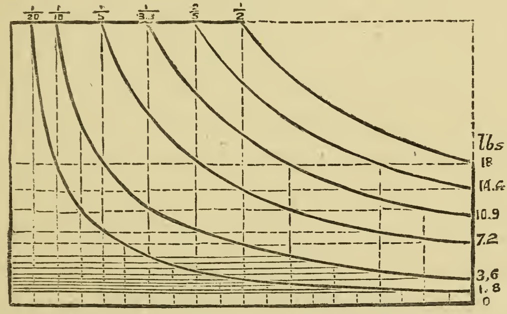
The main purpose for the theoretical diagram has now been fully illustrated; which, as we have stated before, is, "a definition of the curve of expansion:" for example, if the curves in Figs. 14 and 18 are compared, a difference is apparent, although the pressures indicated are alike. Next compare Figs. 15 and 19, and a similar relation occurs with Figs. 14 and 16, also with Figs 15 and 17.

It must be understood that the lengths of the diagrams depicted as S, or termed as equal to the stroke of the piston, include also the clearance and the cubical contents of the steam passage

from the end of the cylinder to the slide-valve facing, and that unless this is added *to the* stroke, the theoretical diagram will be indefinite.

As the comparisons of indicator diagrams are not only interesting, but also instructive, we illustrate a series in one parallelogram by Fig. 20.

Fig. 20.



Comparative Theoretical Indicator Diagram illustrating Several Points of Cut-off.

These are all formed on the Mariotte law, and the vertical and horizontal dotted lines illustrate the points of intersections through which each curve is traced. The pressures at the end of the diagram are known from the formula =

$$\frac{\text{initial pressure}}{\text{grade of expansion}};$$

for example, the first curve is commenced at $\frac{1}{20}$ of the total length; and as the height is 36, the pressure at the other extremity

$= \dfrac{36}{20} = 1{\cdot}8$ lbs. on the square inch at the termination of expansion; then, as the point $\frac{1}{10}$ is *twice* the distance of $\frac{1}{20}$ from the beginning, the pressure of the steam at the other end is *twice* of that below it, or 3·6 lbs; and as $\frac{1}{5}$ curve is in the same ratio, or double of $\frac{1}{10}$, the pressure above it is doubled also, or 7·2, and so on by the same formula the other pressures are obtained; indeed, the formation of the curves depends entirely on the proportion of the length of the supply line to the entire length of the diagram, as we have already stated.

Our next consideration is, what is the best proportion for an indicator diagram? for we have condemned the method of considering the diameter of the cylinder as any function of the matter, because, as we have already explained, the height of the diagram represents the pressure only, and *not* any other limit. The next way to arrive at the solution of this question is to reason about the facts on which it bears.

First, the forward stroke of the piston is theoretically supply steam and its expansion; and the backward stroke, exhaustion and compression on the same side of the piston. This, then, is what the indicator diagram shows above and

below the atmospheric line; but if the engine is non-condensing, of course these features are indicated above that line, as shown by Fig. 20, where the lowest pressure is 1·8 lbs. and the base line zero.

Secondly, the speed and travel of the pencil is due to the pressure of the steam and motion of the barrel. Then, if the pressure shifts or holds the pencil, the longer the act of tracing or marking occurs the more time is given for the operation. We are not forgetting here, either that the speed of the paper will be greater with a large than a small barrel, but remembering that with the greater size there must be a *longer* travel for the pencil or marker, and thus the undulations of the indicator piston will be more truthfully illustrated than with a lesser travel. The matter indeed is really very simple if its aspect is properly viewed; for, after all, it is but a pencil either shifting up and down or still on the paper, and *the more lineal travel given for the pencil the better will its motion be illustrated.* For example, if the indicating piston is jerky in its movement, we require the cause of it; then, if the pencil's travel is short, the indication is reduced to the lowest ebb, instead of being fully and properly depicted, as it would be if the travel were longer.

The *length* of the diagram should therefore bear some proportion to the *velocity* of the piston in its strokes per minute; but this ratio cannot be constant, because the speed of the piston in feet is in proportion to the length of its stroke and the velocity, while the length of the diagram does not take the speed in feet into consideration. We have therefore arranged this matter for future practice as follows, which will doubtless be recognized as an improvement:—

No. of strokes of the piston per minute	100	200	300	400	500	600.
Lengths of the indicator diagrams in inches .	5	6	7	8	9	10.
Diameters of the paper-barrels in inches .	$2\frac{1}{4}$	$2\frac{9}{16}$	$2\frac{7}{8}$	$3\frac{3}{16}$	$3\frac{9}{16}$	$3\frac{15}{16}$.

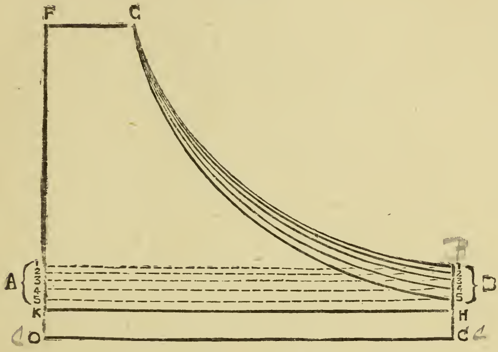
Professor Rankine's Explanation of Theoretical Steam Indicator Diagrams.

In Fig. 21 let A F G B H K represent the indicator diagram of any steam-engine, F being the point of admission, G that of cut-off, B the point of release, H the end of the forward stroke, and K the point where cushioning (if any) begins.

Let the horizontal line through C be the zero line of absolute pressures, so that heights above that line represent absolute pressures of the

steam; B C, for example, being the absolute pressure at the instant of release.

Fig. 21.



Theoretical Indicator Diagram by Professor Rankine.

Through B draw B A parallel to the zero line; and, if necessary, set back the point A, so as to allow for clearance, in order that the length A B may represent the whole volume of steam contained in the cylinder and ports at the instant of release. From A let fall the perpendicular A 0 upon the zero line. Then horizontal distances on the diagram from the vertical line 0 A F represent volumes occupied by the steam in the cylinder.

In expressing the pressures and volumes of the steam such units ought to be adopted that the

product of a pressure and volume may be a certain number of units of work — say, foot-pounds. For eaxmple, if volumes are expressed in *cubic feet*, pressures should be expressed in pounds *on the square foot*; being 144 times the corresponding pressures in pounds on the square inch. If pressures are expressed in pounds *on the square inch*, volumes should be expressed in prisms *one foot long by one inch square*; the volume of such a prism being $\frac{1}{144}$ of a cubic foot.

In the Fig. 21, G 1 represents an expansion curve for *steam gas in a non-conducting cylinder*; the absolute pressure varying inversely as the thirteenth power of the tenth root of the volume, or nearly so.

G 2 represents an expansion curve for *saturated steam, dry on its admission, in a non-conducting cylinder*; the absolute pressure varying inversely as the tenth power of the ninth root of the volume, or nearly so.

Neither of the preceding cases can be perfectly realized in practice; and, therefore, they only represent limits which practical results may approach but not attain.

G 3 represents an expansion curve for *saturated steam, dry on its admission and during its expansion*, being prevented from partially liquefying

by means of a supply of heat through the cylinder. The absolute pressure varies inversely as the seventeenth power of the sixteenth root of the volume, or nearly so.

This is, probably, the best result actually attainable in practice with steam that is not superheated.

G 4 represents a common hyperbola, the pressure varying inversely as the volume simply. To produce such a curve the steam must contain a little liquid water on admission, or immediately afterwards, and that water must be evaporated during the expansion by heat drawn from the cylinder. This is the form of diagram on which calculations are most commonly based, and differs but little from the preceding.

G 5 represents an expansion curve, in which the pressure varies inversely as the fifteenth power of the sixteenth root of the volume; being assumed as an example of the effect of the presence of a still greater quantity of liquid water during the admission, which is evaporated during the expansion.

Then if we calculate in a series of particular cases by the exact formulæ of thermodynamics, a quantity which may be called the *heat of release*, consisting of the total heat, sensible and latent,

of the volume of steam A B at the absolute pressure C B, together with the quantity of heat which that steam would carry off from the cylinder and valve ports, supposing it to expand down to the back pressure without liquefaction, that quantity is found to be given approximately to the accuracy of about 1 per cent. by the following rule:—

Rule for the Heat of Release.

Multiply the product of the absolute pressure and volume of the steam at the point of release by 16 for a condensing engine, or by 15 for a non-condensing engine. The result will be the mechanical equivalent of the heat of release nearly.

(This may, if desired, be reduced to ordinary thermal units by dividing by Joule's equivalent, viz., for foot-pounds and Fahrenheit's scale, 772; for foot-pounds and the centigrade scale, 1390; kilogrammes and the centigrade scale, 424.)

To represent the preceding rule graphically, in Fig. 21, produce AB to D, making AD = 16 AB for a condensing engine, or 15 A B for a non-condensing engine; complete the rectangle A D E 0; then, inasmuch as the area of the rectangle A B C 0 represents the product of the

absolute pressure B C and volume A B of the steam at release, the area of the rectangle ADE0 (= 16 or 15 A B · B C) represents the heat of release in units of work.

The area A B H K of that part of the steam diagram which lies below the pressure of release represents a portion of heat saved out of the heat of release by conversion into mechanical work; and the area A F G B of that part of the steam diagram which lies above the pressure of release represents an additional expenditure of heat, all of which is converted into work. Hence the following rules:—

Rules for the Expenditure and Disposal of the Heat (in Units of Work).

Whole heat expended on the steam = area A D E 0 + area A F G B.

Heat converted into mechanical work = area A F G B H K = area A F G B + area A B H K.

Heat rejected with the exhaust steam = area A D E 0 − area A B H K.

Rule for the Efficiency of the Steam.

Efficiency of the steam =

$$\frac{\text{area A F G B H K}}{\text{area A D E 0} + \text{area A F G B}}.$$

Rules as to Calculation of Theoretical Steam Diagrams.

Supposing the absolute pressure of admission given, the absolute pressure of release is easily calculated for the common hyperbola G4, by dividing the absolute pressure of admission by the ratio of expansion. When the expansion curve deviates but little from a common hyperbola (as G2, G3, and G5), it is often more convenient to take as a first approximation the pressure of release as calculated for a common hyperbola, and then to subtract (for such curves as G2, or G3) or add (for such curves as G5) a correction computed as follows:—Multiply together the approximate pressure of release, the hyperbolic logarithm of the ratio of expansion, and the fraction by which the index of the power of the density to which the pressure is proportional exceeds or falls short of unity. For G2 the index exceeds unity by $\frac{1}{9}$; for G3, by $\frac{1}{10}$; for G5, it falls short of unity by $\frac{1}{10}$.

When the pressure of release is known by observation on an actual diagram, to calculate the index of a curve of the hyperbolic class which approximates to the actual expansion curve; from the difference of the logarithms of the pressures of admission and release subtract the difference

of the logarithms of the volumes of admission and release; the remainder will be the index required.

Should the only table of logarithms at hand be one containing hyperbolic logarithms of rates of expansion, the same question may be approximately solved as follows:—Divide the absolute pressure of admission by the ratio of expansion; then, with the quotient as a divisor, divide the actual absolute pressure of release, and take the difference between the new quotient and 1. Divide that difference by the hyperbolic logarithm of the ratio of expansion: the quotient will be the fraction by which the required index exceeds or falls short of 1, according as the pressure diminishes faster or slower than the density. For example, suppose absolute pressure of admission = 40, ratio of expansion 5, actual absolute pressure of release 8·848. Then $40 \div 5 = 8$; $\frac{8·848}{8} = 1·106$; hyp. log. $5 = 1·609$; $\frac{0·106}{1·609} = \frac{1}{16}$, which is to be subtracted from 1, because the pressure falls slower than the density; finally, $1 - \frac{1}{16} = \frac{15}{16}$, index required.

The area A F G B, above the pressure of release, when the expansion curve is the common hyperbola G4, can be found either by the well-

known formula, absolute pressure of admission × volume of admission × hyperbolic logarithm of the ratio of expansion, or by drawing the curve accurately to scale (which is very easy) and measuring the area by ordinates or by the planimeter.

The corresponding area for other expansion curves is found by formulæ which are given in a previous note. But when the expansion curve deviates but little from a common hyperbola, it is often more convenient to take as a first approximation the area A F G B as for a common hyperbola; then to calculate a correction as follows:—take the difference between unity, and half the hyperbolic logarithm of the ratio of expansion; multiply that difference by the fraction by which the index of the power of the density to which the pressure is proportional differs from unity, and by the approximate area; and then to apply that correction as follows:—

	Hyp. Log. of Expansion.	
	Less than 2.	Greater than 2.
For such curves as G2, G3, .	add.	subtract.
" " G5, ...	subtract.	add.

If the hyperbolic logarithm is 2, no correction is needed.

For example, suppose absolute pressure of admission, 40; volume of admission, 25; ratio of expansion, 5; index, $\frac{15}{16}$; then because hyp. log. $5 = 1{\cdot}609$, we have as a first approximation to the area AFGB, $40 \times 25 \times 1{\cdot}609 = 1609$. Then to compute the correction we have $1 - \frac{1{\cdot}609}{2} = 0{\cdot}1955$; and $0{\cdot}1955 \times \frac{1}{16} \times 1609 = 20$, correction required; and this is to be subtracted; therefore $1609 - 20 = 1589$, corrected area. (By the more exact process the area is 1587, the difference being practically inconsiderable.)

The area of the part of the diagram A B H K that lies below the pressure of release is in every case the product of the volume at release, A B, and the excess of the absolute pressure at release, B C, above the absolute back-pressure, H C.—Appeared in the *Engineer*, Vol. 21.

CHAPTER V.

THE PRACTICAL GEOMETRY OF THE INDICATOR DIAGRAM.

HAVING in the preceding chapters described the arrangement of the indicator, how to use it properly, and the facts to be learnt from that operation, we now enter on the test of the veracity of an indicator diagram; for it is obvious that that figure can be subjected to various methods of production: it may be so imperfectly taken that it does *not* indicate the action of the steam or imperfection of the valve and gear, while, on the other hand, it may be *cooked* or shaped to suit the mind of the operator, or please the eye of the critic, so much so that it is made too perfect, or, what nature has failed in, art has supplied. To guard against all this, then, we must resort to a practical geometrical test, which cannot fail to show the truth, let it be what it may.

As we shall make use of the technical terms "lead, full-steam, and cut-off," in our description of the present matter, we will commence with an explanation of their meaning.

Lead is the position of the slide-valve to allow

a certain amount of steam to enter the cylinder before the piston completes its stroke, so that the motion of the valve is said to be in advance of that of the piston on the return stroke of the latter.

Full-Steam is the position of the valve when it is at its full stroke, and the piston continuing its motion in the same direction as before that.

Cut-off is the position of the valve when it has returned and closed the supply-port or opening, and the piston is advancing as before, but the valve moving in an opposite direction.

We know, therefore, that the slide-valve regulates the flow of the steam into and from the cylinder, and consequently the proportions of the valve, ports, and laps must determine the form of the indicator diagram at certain points.

We must next consider the mechanical action of the valve, which is this:—When the piston of an engine steam-cylinder is at its full stroke, the slide-valve has a lead; or its position is so that the supply-port in the cylinder is opened for a given width, and the steam, therefore, is admitted before the piston has completed its stroke. The use of the lead is to provide for what is termed the concussive action of the piston, and act as a spring, and, therefore, when a slide-valve has no

lead on the piston, the tremor in the cylinder ends betrays the fact; so far is this certain that even, with considerable clearance, with a quick-speed engine the covers rise and fall perceptibly at the centre. Next, then, presume the valve to have opened the supply-port for full steam, it is now at the end of its stroke, but returns and closes the supply-port while the piston is still moving in the same direction as before. The steam is here cut off from the cylinder, and the steam in the boiler is allowed to accumulate for a time —this is the advantage of an early cut-off: time is allowed for evaporation in the boiler, and a saving of steam results by using the smallest quantity possible during each stroke of the piston, and producing sufficient power from expansion. Suppose now, for example, an engine has a 3 ft. stroke of piston, and the valve cuts off when the piston reaches nine inches, or one-fourth, the cylinder is filled to one-fourth of its capacity, or an amount of steam to that extent is taken from the boiler at each first quarter-stroke of the piston. Again, presume the same stroke of piston and diameter of cylinder, but the cut-off to be one-half, or eighteen inches from the end of the stroke, double the capacity of cylinder is open to the boiler—but only about one-half of the

time allowed more than before for the steam to escape from the boiler, although nearly double the amount of the steam is admitted at each first half-stroke of the piston with this latter grade than with the former.

So much for the supply; now for expansion. When the steam is in the cylinder, and the valve has closed the port, the steam is expanding, simply because the piston is advancing, and therefore more space is allotted for the steam. Now as the cut-off is determined by the outer edge of the valve, so is the time of expansion by the inner edge, or on the exhaust side; so that it is absolute that the amount of expansion is due to the time that the valve is traversing over the port, and keeping it closed; therefore the outside lap, or breadth of valve cover, not only settles the cut-off, but also the amount of expansion.

The exhaustion next occurs when the valve has opened the port with its inner edge, and continues until the inner edge of the valve *returns* and closes the port at that end of the cylinder.

Lastly, we have compression, so termed because if there is any steam or air in the cylinder it cannot escape, neither will the vacuum be impaired—perfection of the valve surface to be accepted—and the piston is driving all before it;

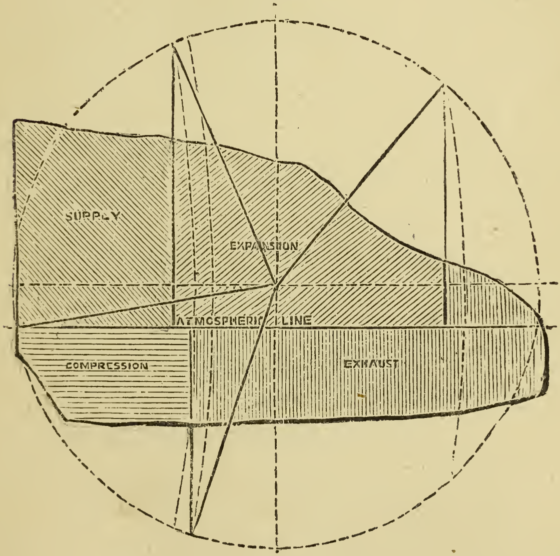
besides, the valve has closed the port until the "lead" occurs, when the release is met by the volume of the steam from the boilers. Evidently, therefore, if the steam is governed by the valve, and the indicator is open to the cylinder, the action of the valve must determine the various points of cut-off, release, and compression on the diagram. But the lengths of the eccentric and main connecting-rods will settle the exact positions of the valve and piston at the various parts of their travel.

Dispensing now with these preliminary observations, we will proceed direct to the geometrical questions and diagrams.

Fig. 22 is a diagram, full size, from the cylinder of a pair of marine engines by an eminent firm in London, to whom we are indebted for the proportions and illustrations here given in relation to this subject. It will be seen that the diagram is partially enclosed within a circle, and this circle is the path of the crank-pin. The diagram is divided into four divisions determined by the action of the slide valve—supply, expansion, exhaustion, and compression. Now the length of the diagram must in all cases represent the stroke of the piston, simply owing to the fact that the figure traced is due to the

pressure of the steam and motion of the indicator card-barrel, and its motion is derived from the steam cylinder piston-rod if possible. Obviously, therefore, the length of the indicator's figure

Fig. 22.



Practical Definition of an Indicator Diagram.

and the piston's stroke are virtually the same; therefore, by dividing the length of the diagram into parts equal to the stroke of the piston in inches, and constructing from that scale a drawing of the slide valve, piston's depth, cylinder,

ports, length of eccentric, and main connecting-rods, a relative proportion of the various positions of the whole of these details in connection with this matter is produced.

The next question is, how the diagram, Fig. 22, was proportionately and truthfully divided into the four parts as shown, and the angles of the crank determined? In order to fully demonstrate the answer to this, we have illustrated the entire matter at a large scale on the folding plate, entitled, "Geometrical Tests of Indicator Diagrams."

Fig. 2 in the plate is an illustration of the cylinder ports, slide valve, position of the crank, the piston at full stroke, and the angle of the eccentric proper to the lead required.

Here let it be noticed that as the eccentric's angle is due to the present position of the crank, it is virtually connected to the crank by a chord of the arc of the eccentric circle, and that there can be no alteration in the length of this chord afterwards; and that in this case the angle of the crank is determined from the angle of the eccentric, and the position of the latter from the slide valve.

The diagram Fig. 3, outside the crank path, is the same as Fig. 22, in page 79, but reduced by

the scale common to the whole of these details. The atmospheric line is below the centre line of the circle or intersects at the angle, L, the crank would be at when the valve commenced the lead. Now assume that the motion has been given to the indicator and the atmospheric line described, the steam admitted to the cylinder—common to the lead—drives the indicator piston from the atmospheric line to A, which in the present case is the greatest pressure.

Fig. 4 now demands attention. Here it is seen that the crank has risen in the direction of the arrow, and the piston has advanced in a relative distance, while the slide-valve has closed the steam-port, and has a retrograde motion to that in Fig. 2. Notice also that the space filled in in the diagram Fig. 5 is the same distance the crank-pin has travelled on the plane line; the dotted curve line in each case shows the relative position of the piston. When the crank arrived at B, the steam was cut off by the slide-valve and expansion commenced, and the non-escape from the cylinder was continued until the valve and crank were in the positions, as shown by Fig. 6.

Turning again to Fig 2, it will be apparent, on comparing it with Fig. 6, what distance the

piston has travelled before the steam is dispensed with or exhaustion commences, also the portion of the circle described by the crank-pin during the same time.

Fig. 7 represents the amount of pressure in the cylinder at the moment the exhaustion occurs; it explains, also, the decrease of pressure from the point of cut-off to the release. The relative diagram areas of supply and expansion are manifest, and the length on the atmospheric line agrees with the plane travel of the crank-pin. Now the *time* allowed for exhaustion is due to the *travel of the slide-valve, less the inside laps*, and while the valve is performing this the piston also completes its stroke and has a return motion. The Fig. 8 will assist now. Here the valve has just closed the supply-ports, terminated exhaustion, and the crank-pin has travelled from C in Fig. 6 to D in Fig. 8, where the relative positions of the pistons are also shown.

Fig. 9 shows the amount of vacuum attained, and that when the crank left C in Fig. 7 the pressure in the cylinder gradually fell to that of the atmosphere before the stroke of the piston was complete, as shown by the pencil's trace intersecting with the atmospheric line. On

reaching this point the pencil descended. When the piston completed its stroke, simultaneously the indicator drum, or barrel, stopped, and moved again in the contrary direction with the piston. The steam was nearly, if not finally, exhausted just after the round corner was formed by the pencil, and the vacuum attained was faithfully recorded, as shown by the line nearly parallel with the atmospheric line. Exhaustion was continued, for the supply-port was open to the condenser, until the piston reached the position depicted in Fig. 8, when the valve closed the port and compression commenced. The motions of the piston and valve being in reverse directions, as indicated by the arrows, no steam or vacuum affected the travel of the piston until the crank reached L in Fig. 11.

Then the steam rushed into the cylinder, simultaneously into the indicator, when the piston of the latter instantly rose, and the pencil ascended to the atmospheric line, which completed the indicator diagram. It will be noticed, however, in the diagram, Fig. 9, that the rounding of the corner must have been formed before the steam, admitted by the lead of the valve, could have caused it, as shown by the angle of the crank at L in Fig. 11. This was due to com-

pression, or, perhaps, leakage of the valve before the port was opened.

We may mention here, in passing, that as the lead of the valve governs the grade of expansion, so that if the lead were increased, the grade of expansion would be reduced.

The diagram, Fig. 9, just explained was taken from the back end of the cylinder, and as all cylinders should be indicated at both ends, the diagram, Fig. 11, was taken also. Fig. 10 shows the position cf the crank, slide-valve, and piston exactly opposite to that in Fig. 2. Fig. 12 depicts the positions of the valve, piston, and crank at the point of cut-off, or reverse to Fig. 4. Here must be noticed the relative position of the pistons, also that although the leads for the valve are alike, the points of cut-off are not the same.

It may be added, as a guide for young engineers, that if equal lead is preserved at each end of the cylinder, unequal expansion results, and *vice versâ*, so that if expansion is equal, uneven outside lap and lead must ensue to obtain the same when the connecting and eccentric rods are on the same side of the crank shaft.

Alluding next to Fig. 14, the relative positions for the termination of exhaustion are illustrated analogous with Fig. 6, common to the opposite

end of the cylinder. Observe the different positions—due to the obliquity of the connecting-rod—of the pistons in each case from the vertical centre lines of the cylinders. The length of the expansion sections in Figs. 7 and 15 will also be found to be unequal in consequence of the action resulting from the length of the connecting-rod. Fig. 16 shows the valve cutting off the communication from the condenser, or the commencement of compression, but the position of the piston in this case, and in Fig. 8, are strikingly at variance, owing to the relative angles of the cranks being on opposite sides of the centres of the crank circles. The length of the compression section on the diagram, Fig. 17, is much shorter also than in Fig. 9.

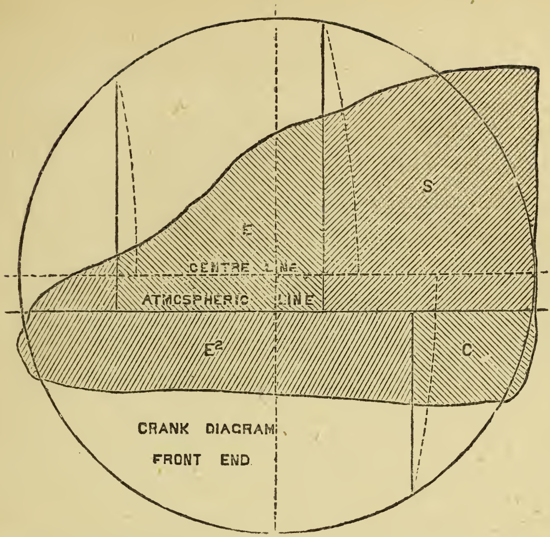
The evidence certain from the preceding remarks is that the diagrams under notice bear a strict relation to the angles of the crank at the points of supply, expansion, exhaustion, and compression. There are, however, other matters to be considered, to which we now direct attention, which are, that indicator diagrams are of two classes, independent of high and low pressure steam, or condensing and non-condensing engines —viz., crank diagrams and piston diagrams.

Our next proceeding is therefore to define the

difference in the two figures, and their separate relations to the crank and the piston. If we notice the illustrations in the plate again, in reference to this, we shall see that all the indicator diagrams are divided by perpendicular lines projected from the intersections of the angles of the crank with its path or circle, which divisions of course refer specially to the angles of the crank at the points of cut-off, expansion, exhaustion, and compression. Now this mode of subdivision does not define the positions of the piston in relation to those for the crank unless they are set out as illustrated; so that when this is not done the formation of the diagram in relation to the speed of the crank-pin is only shown.

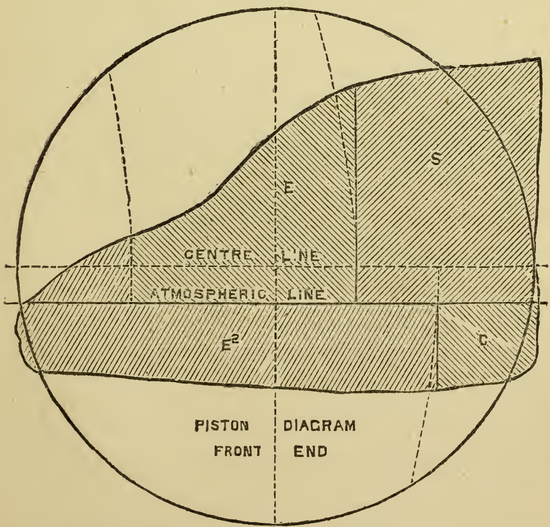
To render a comparison obvious, the illustration Fig. 23 is introduced, which being full size, shows at once the proportions of the divisions of the diagram, taken from the front end of the cylinder, as shown in the plate; the intersection of the vertical lines with the circle depict the positions of the crank-pin during the four stages of the formation of the figure; the dotted curves starting from those points and ending at the atmospheric line, are formed with the length of the connecting rod as the radius, and show thereby the lineal advance of the crank-pin over that of

Fig. 23.



Indicator Diagram taken from the Cylinder shown in the Plate. Full size.

Fig. 24.



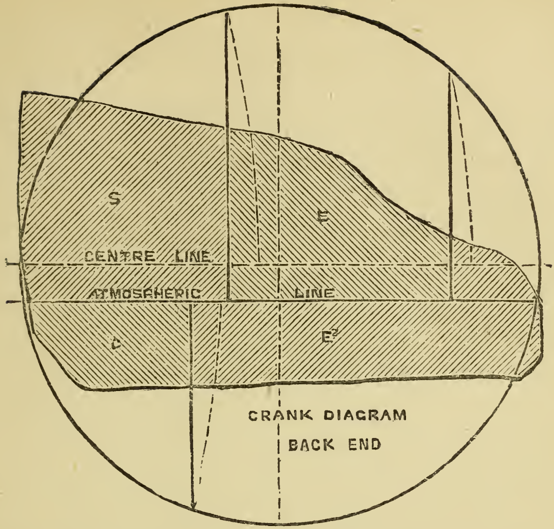
Indicator Diagram taken from the Cylinder shown in the Plate. Full size.

the piston, which is equal to the versed sines of the dotted arcs. The separate relation of this diagram to the piston is shown by Fig. 24, where the divisions contrast with those in Fig. 23. In the present case the vertical lines are tangents to each arc, instead of chords as in the other instance. The letters refer to the stages of construction; thus S is supply steam; and the length of S in Fig. 23 is longer than that in Fig. 24; E is expansion, and its length is shorter in Fig. 23 than in Fig. 24. Exhaustion is next indicated by E^2, the difference in the lengths in both figures is apparent also; lastly, C, which is compression, is shorter in Fig. 24 than in Fig. 23.

As the diagrams in the plate are taken at each end of the cylinder, we illustrate by Fig. 25 the diagram full size, taken at the back end, which shows also the contrast between the crank positions in it and in Fig. 23. The difference in the piston diagrams is shown by Fig. 26, which is that, in this Fig. the supply S is larger than in Fig. 24, and also compression C, while exhaustion E^2 is about one-fourth shorter above.

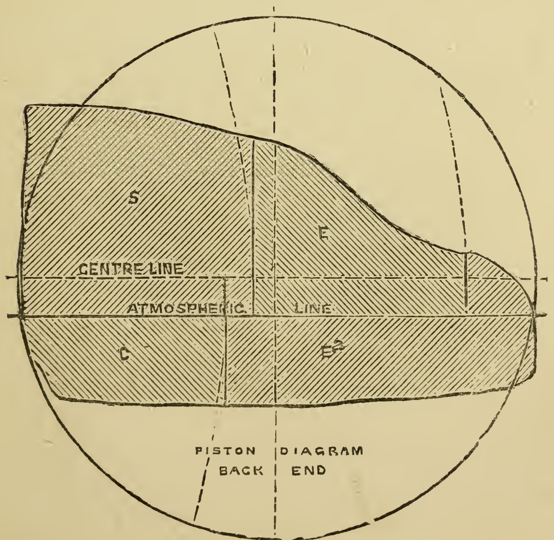
The object attained bythis geometrical test is now apparent, it being that as the actual length of the stroke of the piston is virtually that of the

Fig. 25.



Indicator Diagram taken from the Cylinder shown in the Plate. Full size.

Fig. 26.



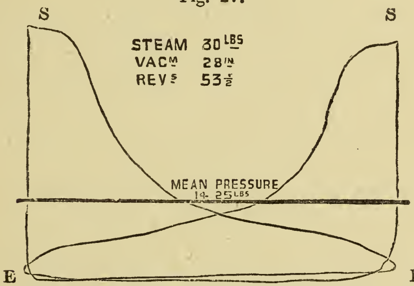
Indicator Diagram taken from the Cylinder shown in the Plate. Full size.

diagram, or *vice versâ*, and the paths of the crank-pin and piston are subdivided virtually also, by the action of the slide-valve we can arrive at a correct conclusion, of not only the proportion of these divisions, but also the *time* occupied to produce each portion of the diagram comparatively; in fact we can see obviously what the steam has been actually performing during the given strokes of the piston, and as the steam is governed by the valve and its gear, we learn also their action. Indeed this geometrical test is the only method that can be adopted to detect fictitious indicator diagrams and the imperfect mechanism of the steam-engine.

CHAPTER VI.

MODERN INDICATOR DIAGRAMS, CONTRIBUTED BY THE MOST EMINENT ENGINEERS IN ENGLAND AND SCOTLAND, TO SHOW THEIR LATEST AND BEST PRACTICE.

Fig. 27.



INVERTED DIRECT-ACTING SCREW-ENGINES.

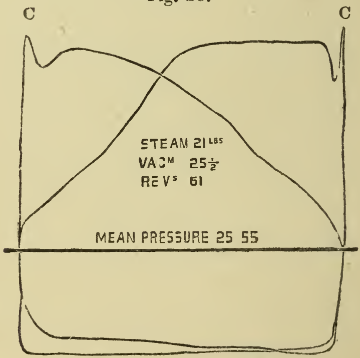
Messrs. Maudslay's indicator diagrams, showing a high *working* grade of cut-off, with *almost* equal lead, and equal expansion. Original scale $\frac{1}{15}$ in. = 1 lb. Half size.

SCREW-ENGINES.—The pair of diagrams shown by Fig. 27 have been very lately taken from both ends of one of the cylinders of a pair of horizontal return-acting engines of 120 h.p., fitted by Messrs. Maudslay into the steamship "Rostoff." The slide-valves are the most modern three-ported type, and the link motion of the usual kind of their late practice. It will be seen that the supply-lines S S are nearly equal in

length and form; also that the curve lines extending to E E depict that the expansion is perfect, and the points of exhaustion alike; indeed the figures *in toto* are *pairs*, in the literal meaning of the word. Although the grade of expansion is high, it is the working power for the engines to drive the ship at eight knots per hour.

As a contrast to the preceding example, Messrs. Maudslay have given the figures shown by Fig. 28. At C C there was a little concussive action of

Fig. 28.



INDICATED HORSE POWER 6867

Return-acting Screw-engines.

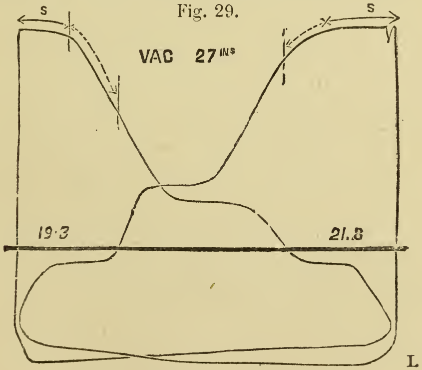
Messrs. Maudslay's indicator diagrams, showing unequal cut-off, *nearly* equal expansion, and equal lead. Original scale, $\frac{1}{10}$ in. = 1 lb. Half size.

the marker, which occurs generally from cylinders whose diameters exceed their lengths. It is produced chiefly by the least priming of the water

in the boiler, or partial condensation of the steam in the cylinder, which compound of steam and water entering the indicator piston causes an excessive undulation, doubtless being the result of the water *striking* the piston *before* the steam acts under it, and the response given by the spring above. The engines these figures were taken from are 1350 h.p., and are fitted into H.M.S. "Agincourt." The indicated horse-power is 6687 collectively.

Leaving the London practice *pro tem.*, we direct next to the diagrams shown by Fig. 29,

Fig. 29.



RETURN-ACTING SCREW-ENGINES.

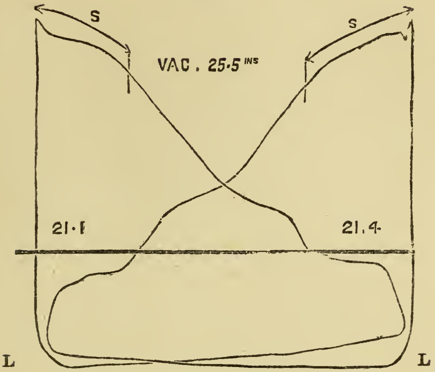
Messrs. Napier's indicator diagrams, showing unequal cut-off, but equal expansion and lead, from aft engine. Original scale $\frac{1}{10}$ in. = 1 lb. Half size.

being the practice of Messrs. R. Napier and Sons, of Glasgow. The forms of both figures are nearly

alike proportionately ; the lengths of the supply lines at S S are unequal, but the actual limit for supply is beyond the straight parts, and a little below on the curve lines ; the cause for the rounding from the straight lines S S to the dotted curves below, is that the steam pressure was reduced in the cylinder, either from partial condensation or contraction of the supply opening. The cause of the undulations of the curves above and below the atmospheric line is either from the shaking of the instrument or oscillation of the steam in the cylinder produced by unsteady expansion and exhaustion ; the lead-corners L L at each end are equal, and show a steady re-admission for the steam.

These diagrams were taken from the aft engine ; we therefore next turn to those taken from the forward engine, depicted by Fig. 30. The supply lines S S are curved, instead of straight, as they should be, owing to the supply of the steam being insufficient to maintain an equal pressure or force in the cylinder during the admission ; or similarly as the result of partial expansion and supply at the same time. The expansion and exhaustion lines above and below the atmospheric line are undulated as those before, and equal lead, L L, also is demonstrated again now.

Fig. 30.



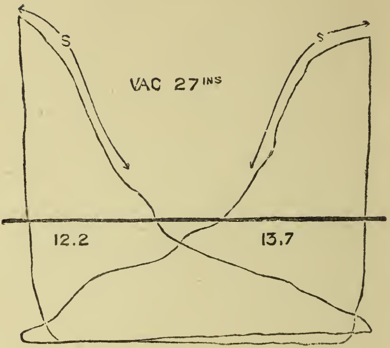
RETURN-ACTING SCREW-ENGINES.

Messrs. Napier's indicator diagrams, showing curved supply lines, undulated expansion lines, but equal lead; from forward engine. Original scale $\frac{1}{10}$ in. = 1 lb. Half size.

The preceding four diagrams are taken at full power; and therefore we know exactly the difference in the mean pressures on the piston on each side—for example, the left-hand diagram in Fig. 29 is 19·3 lbs.; but that in Fig. 30 is 21·1 lbs.; the right-hand diagram in the former figure is 21·8 lbs, but in the latter it is 21·4 lbs. on the square inch.

As a contrast we have introduced the Figs. 31 and 32, to depict the difference in full and half boiler power, and to show also that the supply line depends as much on the *quantity* as the pressure of the steam. These four figures relate

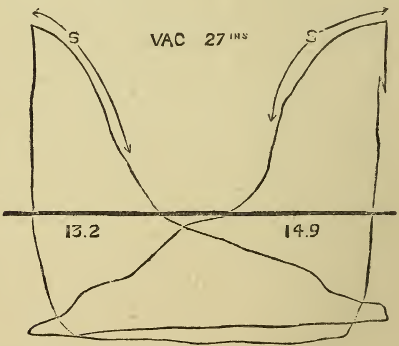
Fig. 31.



RETURN-ACTING SCREW-ENGINES.

Messrs. Napier's indicator diagram, showing the result of half boiler power; aft engine. Original scale $\frac{1}{10}$ in. = 1 lb. Half size.

Fig. 32.

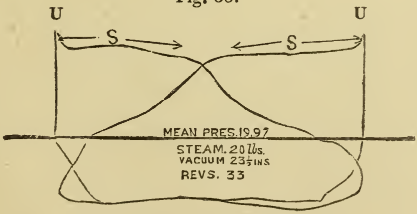


RETURN-ACTING SCREW-ENGINES.

Messrs. Napier's indicator diagram, showing the result of half boiler power; forward engine. Original scale $\frac{1}{10}$ in. = 1 lb. Half size

respectively to those of full power, and the difference in the form and length of the line S is clearly illustrated, it being remembered that it is the horizontal length of S that depicts the point of cut-off; the difference in the mean pressures are also easily understood. All of these diagrams, Figs. 29 to 32 inclusive, were taken from the engines fitted by Messrs. Napier in H.M.T.S. "Malabar," 700 nominal h.p. collectively. The mean pressure of the steam in the cylinder, full power, was 20·9 lbs. on the square inch; vacuum 27 inches; and the revolutions 70 per minute; the indicated horse-power being 4922·31. The mean pressure of the steam at half power in the cylinder was 13·5 lbs.; vacuum, 27 in.; number of revolutions, 57·66, and the indicated horse-power 2619, each result being collective.

Fig. 33.



DIRECT-ACTING SCREW-ENGINES.

Messrs. Watt's indicator diagrams, showing *full* supply steam, even expansion, with equal exhaustion and lead. Original scale $\frac{1}{16}$ in. = 1 lb. Half size.

Leaving the Scottish practice for the present,

and returning to the English examples again, we refer now to the oldest firm—Messrs. James Watt and Co.; their recent and best practice being depicted by Fig. 33.

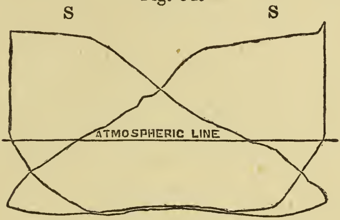
These were taken from the steam-ship "Ipojuca." The engines are direct acting, the crank-pins making 33 revolutions per minute, with 20 lbs. pressure of steam per square inch in the boiler, and the condenser attaining $23\frac{1}{2}$ lbs. of vacuum. The marker was slightly undulated at U U, but the lines S S are creditably straight, and parallel with the atmospheric line.

We may here add in passing that as the lines S S are the indications of the *stationary* positions of the pencil or marker when the indicator piston is *held up* by the steam pressure, these lines *should* be always parallel with the atmospheric line, because the marker is stationary then *also* when describing it.

To show that the least difference in the working of the engine is faithfully indicated by the indicator diagram, Fig. 34 is introduced. Here the supply lines S S show that the pressure of the steam gradually decreased after it entered the cylinder, instead of, as in Fig. 33, maintaining it until the cut-off occurred. In this case the steam pressure in the boiler was 22 lbs.; vacuum 23·5

inches in the condenser, and revolutions 36; the indicated mean total pressure on the piston was 19·15 lbs., and in Fig. 33, 19·7 lbs.

Fig. 34.

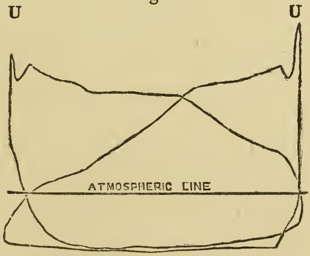


DIRECT-ACTING SCREW-ENGINES.

Messrs. Watt's indicator diagrams, showing a slight deflection of the lines S S Original scale $\frac{1}{16}$ in. = 1 lb. Half size.

Another example by this firm is shown by Fig. 35. Here the indicating marker undulated

Fig. 35.



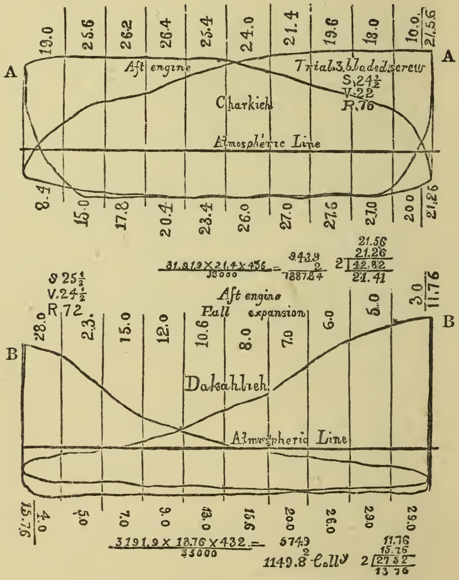
DIRECT-ACTING SCREW-ENGINES.

Messrs. Watt's indicator diagrams, showing vertical undulation at the commencement of the supply steam line. Original scale $\frac{1}{16}$ in. = 1 lb. Half size.

at U U, and from thence gradually lowered until the cut-off occurred. These diagrams were taken from one of the cylinders of the engines fitted

into the steam-ship "Medusa," twin screws*—the starboard aft cylinder; steam pressure in the boiler 25 lbs.; vacuum 24 inches, and revolutions 124; mean indicated pressure on the piston, 21·5

Fig. 36.



Return-acting Screw-engines.

Messsrs. Rennie's indicator diagrams, from their best return-acting screw engines, A A being full supply steam, and B B expansion. About half size.

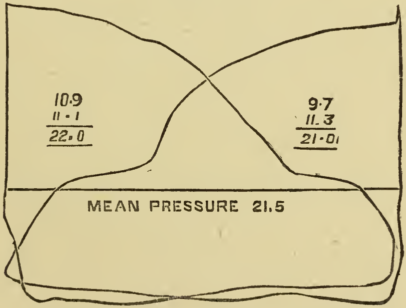
The diagrams shown by Fig. 36 are from

* For full particulars of engines, see "Modern Marine Engineering," also Messrs. Rennie's.

engines of 350 h.p., collectively, fitted lately by Messrs. Rennie into the Egyptian steam-ships "Charkieh," and "Dakahlieh." The two pairs of figures show clearly the results of working with full steam at half-stroke and an earlier cut-off by a separate valve.

As the engines for transport service should be reliable, we illustrate next the indicator diagrams of engines fitted by Messrs. Laird, of Birkenhead, into H.M.T.S. "Euphrates," sister ship to the

Fig. 37.

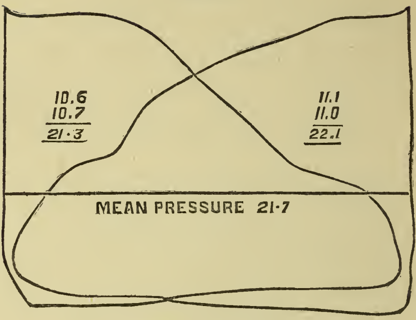


RETURN-ACTING SCREW-ENGINES.
Messrs. Laird's indicator diagrams, showing almost equal supply and exhaustion lines ; forward engine. Original scale $\frac{1}{8}$ in. = 1 lb. Half size.

"Malabar," by Fig. 37; the first being from the forward cylinder, and the second, Fig. 38, from the aft cylinder.

The engines are 700 h.p. nominal; horizontal return action; diameter of cylinder, 94 in.; stroke

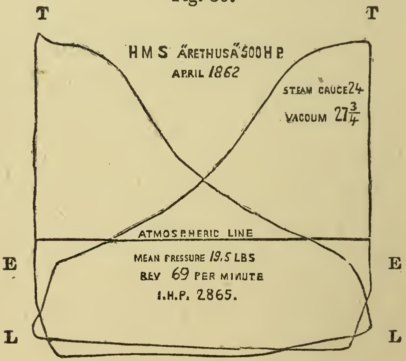
Fig. 38.



Return-acting Screw-engines.

Messrs. Laird's indicator diagrams, showing unequal lines of supply and expansion; aft engine. Original scale $\frac{1}{8}$ in. = 1 lb. Half size.

Fig. 39.



Double-trunk Screw-engines.

Messrs. Penn's indicator diagrams, showing equal supply, expansion, and lead, with nearly equal exhaustion. Half size.

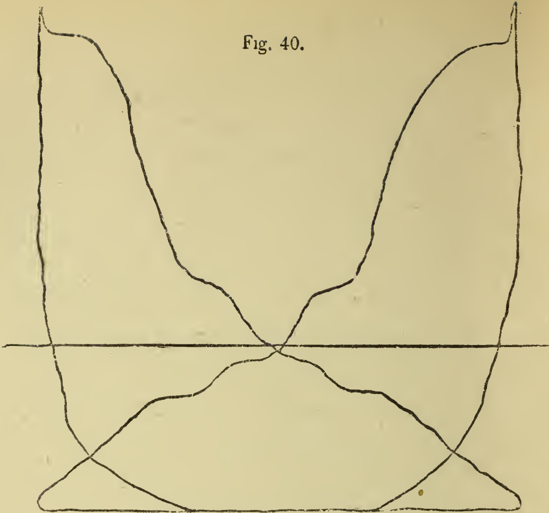
of piston, 4 ft. 6 in.; number of revolutions of crank-pin, 62·4 per minute; pressure of steam in the boiler, 27 lbs.; vacuum in the condenser, 26·5 in.; and the indicated horse-power, 4981.

We now turn to a pair of diagrams that are *really twin*, or nearly duplicate one with the other, as shown by Fig. 39. Here T T are the limits of steam pressure, E E opposite the points of exhaustion, and L L the indications of lead.* As the forms of the lines are explanatory without description, we allude next to Figs. 40 and 41.* These diagrams were taken from one of the cylinders of the three-cylinder arrangement as constructed by Messrs. Maudslay, Sons, and Field.

The slide-valve used is the three-ported kind, and is worked by crank-and-spur gearing in the place of eccentrics and link motion. The diagrams show that the time for *full* supply steam was very short, as the horizontal length of the supply-lines indicate. The actual cut-off was at one fifth of the stroke of the piston; therefore the openings caused by the valves must have been gradually reduced in width and area before half the time expired for the admission of the steam.

* For full particulars of engines, see "Modern Marine Engineering."

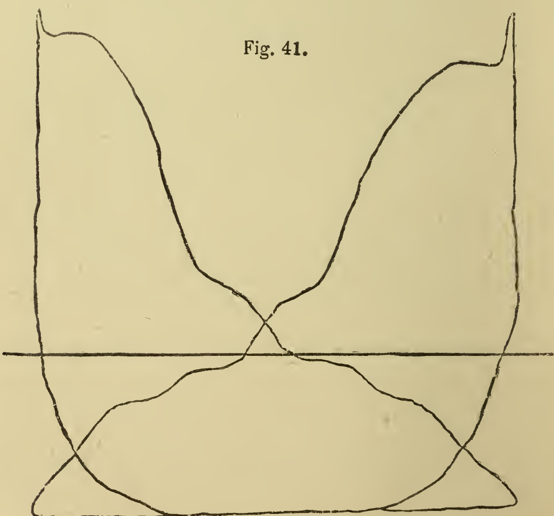
Fig. 40.



RETURN-ACTING SCREW-ENGINES.

Messrs. Maudslay's indicator diagrams, showing nearly equal supply, with equal expansion, exhaustion, and lead; forward engine.

Fig. 41.



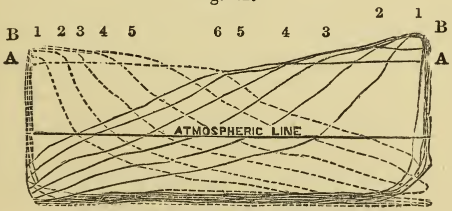
RETURN-ACTING SCREW-ENGINES.

Messrs. Maudslay's indicator diagrams, showing nearly equal supply, with equal expansion, exhaustion, and lead; aft engine.

We may here again remind the young engineer that when the supply-line is inclined or angular a rapid reduction of the pressure of the steam occurs during admission; but when that line is horizontal for the half of its length and terminates suddenly with a curve, as in Fig. 29 in page 93, the reduction of the pressure is slower; but when those lines *are fully* horizontal, as with Fig. 33, in page 97, then the *full* pressure must have been maintained during the time for admission.

Our next contributed example is shown by Fig. 42, which defines the result of using the

Fig. 42.



RETURN-ACTING SCREW-ENGINES.

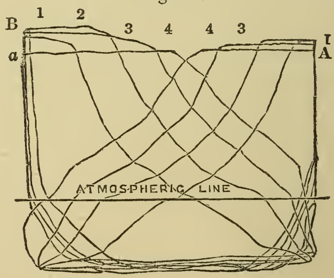
Messrs. Napier's indicator diagrams, howing six grades of cut-off, by the link motion and slide-valve. Original scale $\frac{1}{15}$ lb. to an inch. Half size.

slide-valve and the link motion for the purpose of cutting off the steam at 6 points of the stroke. The points are depicted by the figures 1 to 6 being directly over them. The positions of the letters A A and B B depict that at A A the full

pressure was the lowest at the sixth grade of cut-off, or at half-stroke, and the highest at B B, the first grade, or at $\frac{1}{18}$ of the stroke. These two limits of the pressure show that the readmission of the steam is always more suddenly effective with an early cut-off than otherwise, and that the lead being sooner causes it. These diagrams were taken by Messrs. Napier from the cylinders of the engines fitted by them in the Turkish frigate "Osman Ghazy." The diameter of the cylinders is 92 in.; stroke of the piston, 4 ft.: with full supply steam at the 6th point of cut-off, the pressure was 21 lbs.; vacuum, 26 in.; revolutions 49, and the indicated horse-power, 3782.

Another example of the same class of diagrams is shown by Fig. 43, depicting only four grades

Fig. 43.



Direct-acting Screw-engines.

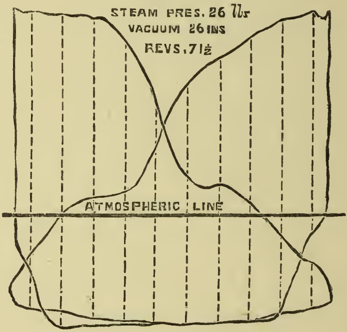
Messrs. Watt's indicator diagrams, showing four grades of expansion, with the link motion and slide-valve. Original scale $\frac{1}{16}$ in. = 1 lb. Half size.

of expansion, and the points of cut-off by the figures 1 to 4. The diagrams on the right hand, A, are almost perfect in form, as the supply lines are horizontal, the expansion lines true curves—particularly those of Nos. 1 and 2—and the exhaustion lines indications of sudden reduction of the pressures, while the lead-lines are in proportion to the degree of cut-off. The left-hand diagrams, B, are not nearly so perfect, except the diagram *a*, which is the longest limit for supply, or No. 4, where the supply line is horizontal, whereas all the others are inclined, showing an uneven pressure; but we may remark that, as these lines are higher from the atmospheric line than those opposite, the mean pressures of both are nearly equal; and as it is the mean pressure that we deal with, for practical purposes the difference in the forms of the supply lines is not of any great importance in this instance.

The diagrams Fig. 44 were taken from the aft cylinder, $104\frac{1}{4}$ in. in diameter, stroke of the piston 4 ft., of the trunk engines, 1000 horse-power nominal, fitted by Messrs. Penn into H.M.S. "Bellerophon." The cylinders are jacketed, the steam is superheated, and the surface condensation used. The diagrams depict, an unequal supply, and the steam slightly throttled, by

the right-hand figure ; the expansion and exhaustion lines are nearly duplicates ; but the lead lines are unequal, the least lead being with the most full supply line.

Fig. 44.



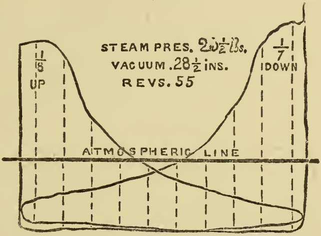
DOUBLE-TRUNK SCREW-ENGINES.

Messrs. Penn's indicator diagrams, showing the result of using the slide-valve and link motion for an early cut-off, termed "expansion by link motion." Original scale, $\frac{1}{10}$ in. = 1 lb. Half size.

The next set of diagrams, shown by Figs. 45 and 46, were taken by Messrs. Penn from a pair of inverted direct-acting engines, of 120 horse-power nominal collectively ; diameter of each cylinder 44 inches, and the stroke of the piston 3 ft. The slide-valve is fitted with a sliding expansion valve on the back, both being worked by link motion. The engines were fitted in the Russian Steam Navigation and Trading Company's steamship "Azof." The diagrams are so

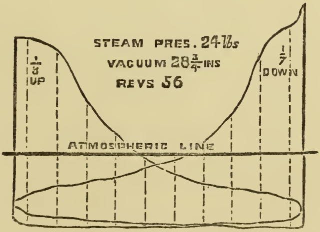
perfect in the expansion lines, with complete exhaustion, that they need no comment, save that the supply lines of the right-hand figure are not as full as those opposite—due to the grade of cut-off being earlier in the former than the latter.

Fig. 45.



INVERTED DIRECT-ACTING SCREW-ENGINES.
Messrs. Penn's indicator diagrams, showing perfect expansion and lead lines; from the aft cylinder. Original scale, $\frac{1}{16}$ in. = 1 lb. Half size.

Fig. 46.



INVERTED DIRECT-ACTING SCREW-ENGINES.
Forward cylinder.

We may add that Messrs. Penn's indicator gear is precisely as what Messrs. Napier use, as illustrated by Fig. 50 in page 113, excepting the loop

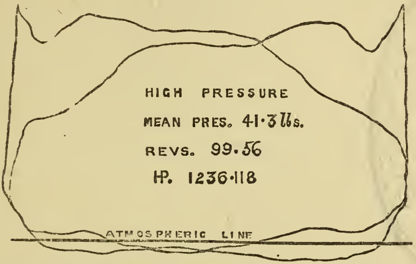
for the vibration of the swing lever, as Messrs. Penn prefer a rod to connect at one end to the lever, and secured by a pin at the other to either the piston or air-pump rods, and Messrs. Maudslay prefer the same.

COMPOUND SCREW-ENGINES.—Messrs Maudslay have lately fitted H.M.S. "Sirius" with compound marine screw-engines of 350 horse-power collectively. The high-pressure cylinder is 34 in. in diameter, and is situated *within* the low-pressure cylinder, whose diameter is 75 in.; the stroke of the piston being 2 ft. 9 in. Pressure of steam in the boilers, 49 lbs.; number of revolutions of the crank-shaft, 99·56 per minute; and total indicated horse-power, 2293·68.

The diagram Fig. 47 is from the high-pressure cylinder, and shows that the steam-supply lines are undulated at the commencement—due to the admission of the steam, or lead, being rather early, as shown by the curves at the lead-corners. The exhaust line is undulated a little also over the atmospheric line; which is caused by the steam not being entirely out of the cylinder on the return stroke of the piston; but taking the figures in the aggregate, they pair well, and show a good result.

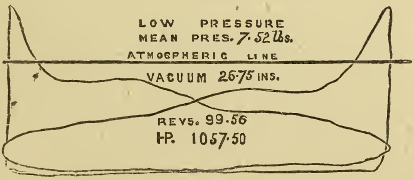
The low-pressure diagrams shown by Fig. 48 pair better than those preceding, and indicate a vacuum of 12 lbs. on the square inch, with a

Fig. 47.



COMPOUND HORIZONTAL RETURN-ACTING SCREW-ENGINES. Messrs. Maudslay's indicator diagrams, showing equal cut-off, expansion, exhaustion, and lead. High pressure; aft cylinder. Original scale, $\frac{1}{20}$ in. = 1 lb. Half size.

Fig. 48.



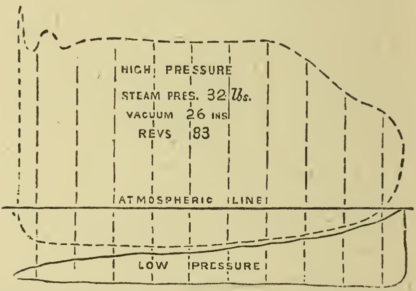
COMPOUND HORIZONTAL RETURN-ACTING SCREW-ENGINES. Messrs. Maudslay's indicator diagrams, showing complete expansion. Low pressure; aft cylinder. Original scale, $\frac{1}{10}$ in. = 1 lb. Half size.

pressure of steam on entering the cylinder 5·5 lbs. on the square inch, which expanded almost theoretically perfect in practice.

The diagrams shown by Fig. 49 were taken

from the same engines when the ship was at "moorings," and illustrate how the high and low-pressure indications "piece" together at proportionate scales.

Fig. 49.

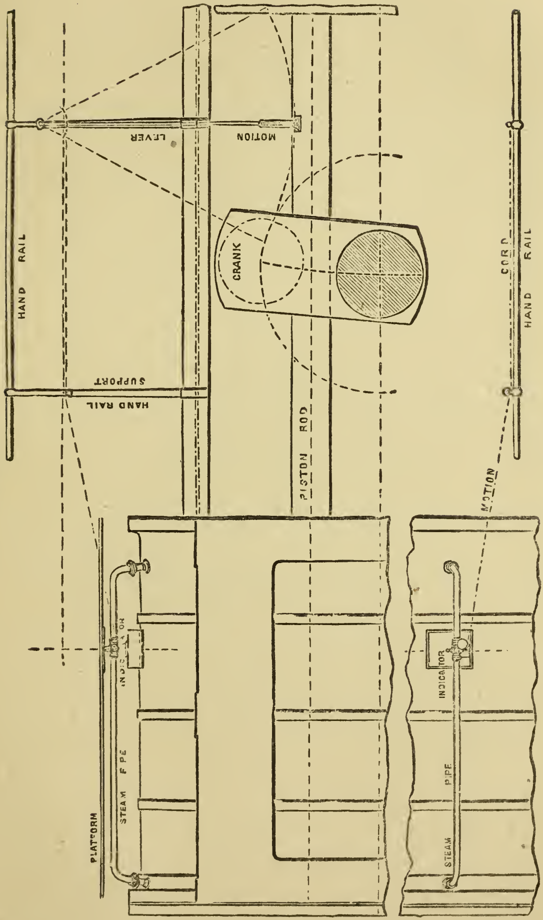


COMPOUND HORIZONTAL RETURN-ACTING SCREW-ENGINES.
Messrs. Maudslay's indicator diagrams "pieced" together. High pressure, half size; low pressure, quarter size in height, and half size in length.

MESSRS. NAPIER'S INDICATOR GEAR FOR RETURN-ACTING HORIZONTAL DOUBLE PISTON-ROD SCREW-ENGINES. — The firm has kindly given us a working drawing of this gear, which we illustrate at a working scale by Figs. 51 and 52 for the benefit of young engineers and students who have yet to learn that the arrangement of the indicator gear should be as simple as possible.

The motion lever is the same kind as illustrated for direct-acting engines in page 18, excepting that in this case the looped lever is above

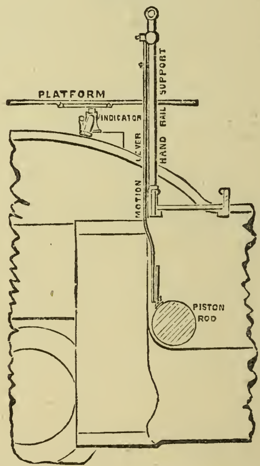
Fig. 50.



Side elevation and plan of Messrs. Napier's indicator gear for return-acting engines. Scale, $\frac{1}{3}$ in. = 1 ft.

the piston-rod instead of below it. The steam pipe is connected to the top side of the cylinder rather than to the ends, and the indicator is situated nearer to one end than the other; also the cord leading from the first motion pulley is at an angle; all of which are faults, although perhaps not of great magnitude.

Fig. 51.

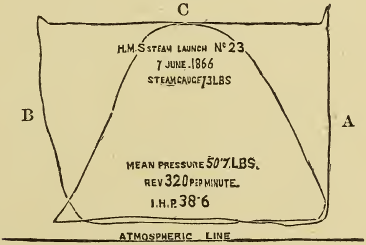


End elevation of Messrs. Napier's indicator gear for return-acting screw-engines. Scale $\frac{1}{3}$ in. = 1 ft.

STEAM-LAUNCH TWIN SCREW-ENGINES.—Messrs. Penn's diagrams, shown by Fig. 52, indicate that the cut-off points, C, are midway of the length of the figures, and that the expansion and exhaustion

lines are nearly equal; but not so the leads; the difference being that the right-hand diagram shows the least amount of lead possible, and a vertical admission line, A; but that at the left hand depicts the lead indication larger, and the admission line, B, at an angle for the most part of its height.

Fig. 52.

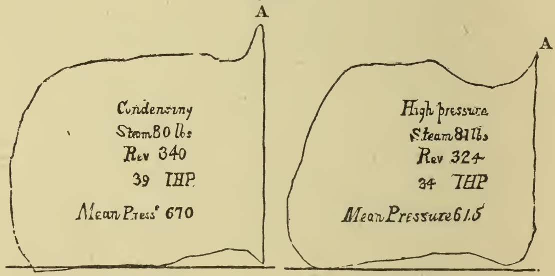


STEAM-LAUNCH TWIN SCREW-ENGINES.
Messrs. Penn's indicator diagrams, showing *full* supply steam and equal expansion, but unequal leads. Half size.

Messrs. Rennie have constructed a great number of steam-launch engines with surface condensers that are arranged with piping and valves so as to convert the engines into high-pressure or condensing while they are in motion. The diagrams shown by Fig. 53 are examples from the same engine at the period of using the steam as described. It will be seen that the condensing diagram is the better of the two, as the supply

line is more regular than the other; but the admission lines, A A, are both nearly equal; while the lead in the left-hand figure is not indicated at all—due of course to the slip of the pencil and paper occurring in reverse directions at the same instant—an evidence also that the person who was taking the figure was careless at that moment.

Fig. 53.



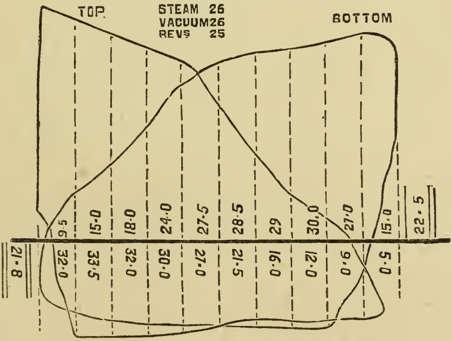
STEAM-LAUNCH TWIN-SCREW ENGINES.
Messrs. Rennie's indicator diagrams, showing the results of condensing and non-condensing the steam with the same engines. Half size.

CHAPTER VII.

INDICATOR DIAGRAMS TAKEN FROM THE MOST IMPROVED MODERN PADDLE-WHEEL ENGINES.

Paddle-wheel engines.—The diagrams shown by Figs. 54 and 55 were taken from the cylinder of the engines constructed by Messrs. J. and G. Rennie, and fitted by them into the P. & O. Co.'s

Fig. 54.



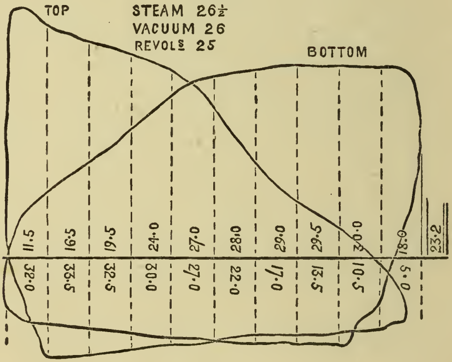
Oscillating Paddle-engines.

Messrs. Rennie's indicator diagrams, showing unequal supply, nearly even expansion and exhaustion, and equal lead; starboard cylinder. Original scale, $\frac{1}{10}$ in. = 1 lb. Half size.

steam-ship "Nyanza," a vessel of mercantile record and renown. On comparing the figures, we

notice that the top diagrams are very like each other at a glance-like observation; but on testing the pressures they show a little difference, which may result from the imperfection of the instrument as much as the action of the steam. With both figures the supply of the steam appears to

Fig. 55.



Oscillating Paddle-engines.

Messrs. Rennie's indicator diagrams, showing unequal supply, nearly even expansion and exhaustion, and equal lead; port cylinder. Original scale, $\frac{1}{10}$ in. = 1 lb. Half size.

have been limited sufficiently to allow the decrease of the pressure in the cylinder from the commencement of the stroke, and therefore the supply lines are inclined as depicted. The lines of expansion are nearly duplicates, and also the exhaustion portion shows an almost equal vacuum direct to the lead-point. The latter indication in

the bottom figures is nearly alike, while in the top figures a little difference is shown; that in Fig. 55 is a moderate curve, but in Fig. 54 it is an irregular line for some distance, and then terminates with a *set-off* to join the full steam line. Now the cause for the formation of that set-off has often been questioned by really veritable authorities. One side attributes it to the action of the steam, and the other to the faulty arrangement of the gear. Now if we remember that the formation of any straight horizontal line must result from the paper *moving* while the pencil is *still*, we can determine that the cause for the set-off is that the pencil must have momentarily stopped before the lead line joined the admission line; and also the barrel must have moved quicker at that instant than before. Of course the barrel only stopped then, but its movement must have been sudden at the conclusion. The cause for this evidently lay with the gear and the steam also; the pencil, it being remembered, is the steam indicant, while the paper represents the gear. Or it might be that the indicator piston stuck fast at the same time that the jerk was given to the barrel. At any rate the error shows that the pencil almost ceased its motion for the time the set-off was described; and the

cause is either that the steam pressure ceased in the indicator from direct condensation, if the piston did not hitch from friction. The jerk given to the barrel proves that the string must have slackened slightly before and become tight at the conclusion. The slackening would be the result of the lever or pin being loose on its bearing.

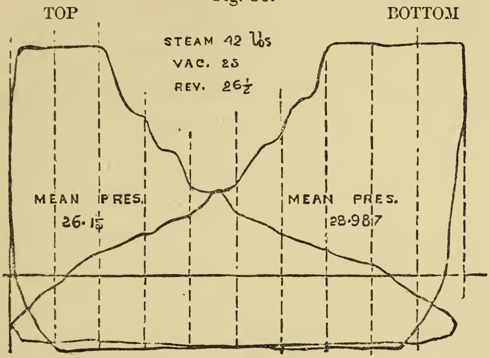
The "Nyanza's" cylinders are 6 ft. $6\frac{7}{16}$ in. in diameter, and the pistons have each a stroke of 7 ft. The engines are 450 nominal horse-power; mean pressure in the cylinders, 22·45; number of revolutions per minute, 25; indicated horse-power, 2304.

The condensation is on the surface-condensing principle; and the diagrams indicate that it has been well carried out.

The next examples, shown by Figs. 56 and 57, are really beautiful specimens of duplication; and the firm—Messrs. Napier—deserves praise for their production; for the two on the right hand are twins, and those on the left are of similar relation; the irregular expansion lines are also alike, so that what caused them in one case occurred in the others. The cause is that the indicator piston undulated during the expansion of the steam, and the undulation results from

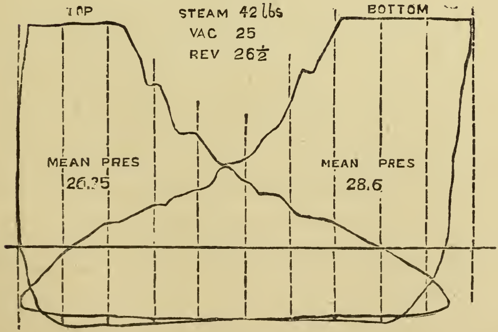
infinite condensation of the steam. The engines, of 210 nominal horse-power, are fitted in the

Fig. 56.



Port cylinder.

Fig. 57.



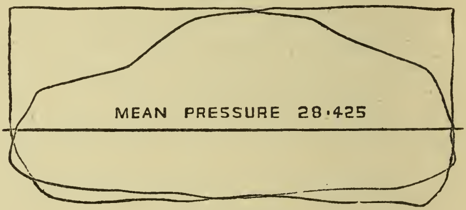
OSCILLATING PADDLE-WHEEL ENGINES.

Messrs. Napier's indicator diagrams, showing *full* steam, equal supply, expansion, exhaustion, and lead; starboard cylinder. Original scale, $\frac{1}{16}$ in. = 1 lb. Half size.

steam-ship "Caroline;" diameter of cylinder, 54 in.; stroke, 6 ft.; mean pressure in cylinders, 27·496; number of revolutions per minute, 26½; indicated horse-power, 1213·69.

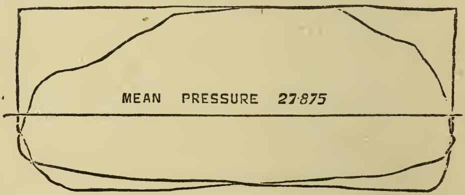
Messrs. Laird's practice next comes under consideration, illustrated first by Figs. 58 and 59.

Fig. 58.



Port cylinder.

Fig. 59.



OSCILLATING PADDLE-WHEEL ENGINES.

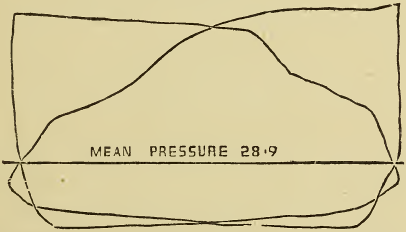
Messrs. Laird's indicator diagrams, showing *full* supply steam, unequal expansion, equal exhaustion and lead; starboard cylinder. Original scale, $\frac{1}{8}$ in. = 1 lb. Half size.

These are taken from a pair of cylinders 59 in. in diameter; stroke of piston, 5 ft. 6 in.; nominal horse-power, 240; steam pressure in boilers, 21·5;

vacuum in condensers, 25·5 in.; revolutions, 24; indicated power of the engines, 1231; fitted into the steam-ship "Guará," for the Brazilian S.S. Co.

Leaving these we notice next a pair of diagrams taken from angular oscillating engines by the same makers, depicted by Fig. 60. The supply steam lines are slightly inclined, which may have resulted from two causes; one being defective

Fig. 60.

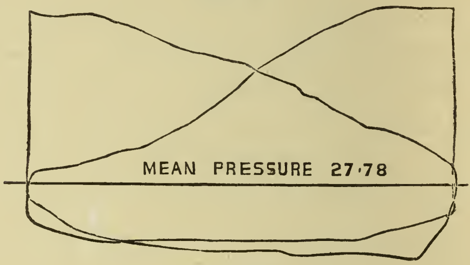


ANGULAR OSCILLATING PADLLE-WHEEL ENGINES.
Messrs. Laird's indicator diagrams, showing even supply, expansion, exhaustion, and lead. Original scale, $\frac{1}{12}$ in. = 1 lb. Half size.

supply or pressure of steam, and the other the contraction of the steam opening, caused by the valve, or what is often termed "throttling the steam." The engines are 260 horse-power nominal; diameter of cylinders, 64 in.; stroke of the piston, 6 ft.; steam pressure in the boilers, 26 lbs.; vacuum, 24 in.; number of revolutions per minute, 30; indicated horse-power, 2028 collectively.

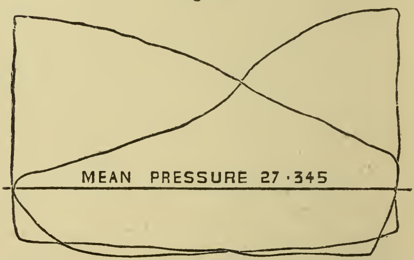
The diagrams shown by Figs. 61 and 62 are also by Messrs. Laird, but are not as creditable as those preceding. The engines are 100 horse-power

Fig. 61.



Port cylinder.

Fig. 62.



Inclined Direct-acting Paddle-wheel Engines.

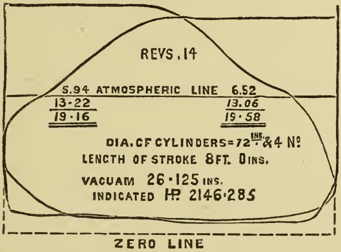
Messrs. Laird's indicator diagrams, showing a contracted supply of steam, uneven expansion, nearly equal exhaustion and lead. Original scale, $\frac{1}{8}$ in. = 1 lb. Half size.

nominal, fitted into a tug-boat. Diameter of cylinders, 38 in.; stroke of the piston, 4 ft. 6 in.; number of revolutions per minute, 39·5; steam,

29 lbs.; vacuum 24 in.; indicated horse-power, 673 collectively.

As a contrast with those preceding, we direct attention to the diagrams shown by Fig. 63, and those shown by Fig. 64, to define the result of

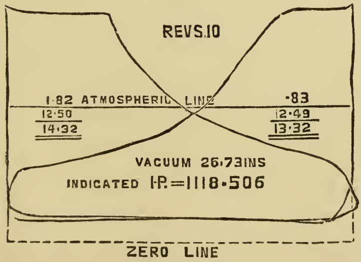
Fig. 63.



VERTICAL RETURN-ACTING PADDLE-WHEEL ENGINES.

Messrs. Maudslay's indicator diagrams, showing *full* supply, unequal expansion and exhaustion, with equal lead; starboard engine. Original scale, $\frac{1}{10}$ in. = 1 lb. Half size.

Fig. 64.

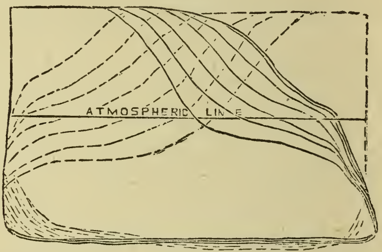


VERTICAL RETURN-ACTING PADDLE-WHEEL ENGINES.

Messrs. Maudslay's indicator diagrams, showing the result of half-boiler power in comparison with Fig. 63; starboard engine. Original scale, $\frac{1}{10}$ in. = 1 lb. Half size.

lower pressure and slower speed. The diagrams in Fig. 63 are taken with full boiler power, but those depicted by Fig. 64 with half boiler power

Fig. 65.

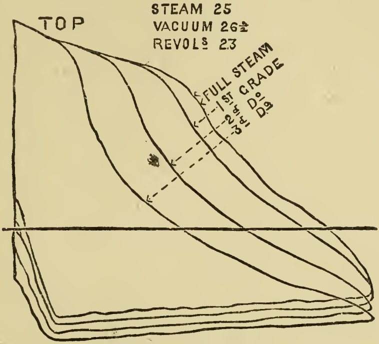


VERTICAL RETURN-ACTING PADDLE-WHEEL ENGINES.
Messrs. Maudslay's indicator diagrams, showing the result of the expansion gear used; starboard engine. Original scale, $\frac{1}{10}$ in. = 1 lb. Half size.

only; and as a contrast to both we illustrate a series taken from the same engines with the cam expansion gear on, shown by Fig. 65. The engines are fitted in H.M.S. "Terrible;" but, although old and costly, show clearly that with a long stroke for the piston, *time* is allowed for the full expenditure of the steam, and that the firm recognized that fact when they designed the engines. We may add that the originals of the above diagrams were taken in March, 1865; the mean indicated full boiler-power for six runs was 2059·684; speed of the ship 10·5 knots per hour, and the slip of the paddle-floats 21·7 per cent.

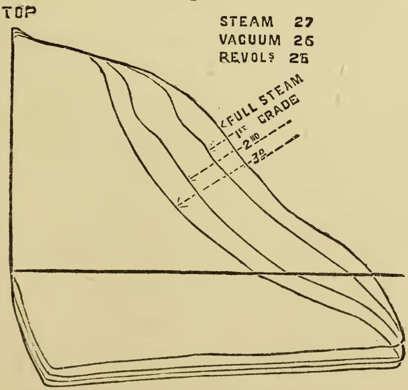
Messrs. Rennie's examples now claim attention, as depicted by Figs. 66 and 67. These diagrams

Fig. 66.



Starboard cylinder.

Fig. 67.



Oscillating Paddle-wheel Engines.

Messrs. Rennie's indicator diagrams, showing the result of using the steam expansively; port cylinder. Original scale, $\frac{1}{10}$ in. = 1 lb.

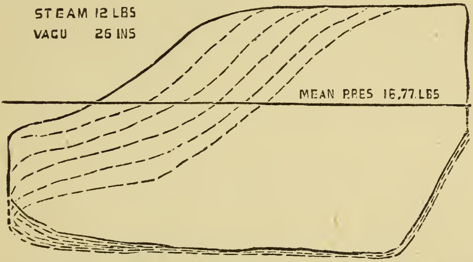
were taken from the same engines as those were that are illustrated in pages 117, 118, by Figs. 54 and 55. In both cases the steam was evidently throttled to some extent; but what was lost in steam pressure was made up with vacuum, as can be seen from the diagrams Figs. 66 and 67, and that when full steam was admitted the vacuum indication was the least, and *vice versâ* with the third grade of cut-off, and also with the intermediate grades respectively.

We learn here, again, that the *time* for the admission of the steam proportionates the *time* for exhaustion; and hence with an early cut-off a better vacuum is attainable than with a later limit for the supply. In fact, the whole matter is regulated by the relation that the *time* bears to the *pressure*; indeed, the indicator diagram is only an evidence of time and pressure, and when their relation is understood the facts are as simple as need be. As we have often stated before, the horizontal lines are "even or full continuous pressures," the vertical lines, "instantaneous pressures," and that *all* deviations from these are *alterations* of pressures; then, as the pressure is to the time, so is the time to the indication.

Another example of a series of expansion diagrams is shown by Fig. 68, taken by Messrs.

Maudslay from the cylinder of one of the engines of 800 horse-power fitted by the firm into the steamship " Oronoco." The diagrams show very

Fig. 68.

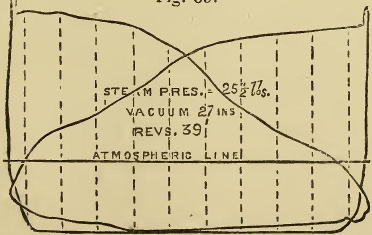


VERTICAL OSCILLATING PADDLE-WHEEL ENGINES.
Messrs. Maudslay's indicator diagrams, showing six grades of cut-off, and nearly equal lead. Original scale, $\frac{1}{8}$ in. = 1 lb. Half size.

clearly the proportion that the supply line bears to the expansion curve, also the relative points of exhaustion and lead.

The diagrams shown by Fig. 69 were taken from one of the cylinders of a pair of inclined di-

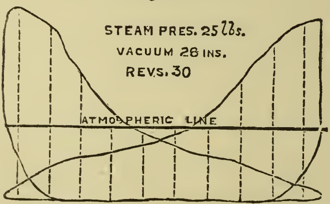
Fig. 69.



DIRECT-ACTING INCLINED PADDLE-WHEEL ENGINES.
Messrs. Penn's indicator diagrams, of nearly duplicate proportions. Original scale, $\frac{1}{16}$ in. = 1 lb. Half size.

rect-acting engines of 100 horse-power collectively, and fitted by Messrs. Penn in an Egyptian tug-boat. Diameter of cylinder, $38\frac{1}{2}$ in.; stroke of piston, 4 ft. 6 in. The figures show very nearly equally full supply-steam lines, and the remainder almost duplicates. The diagrams illustrated by

Fig. 70.

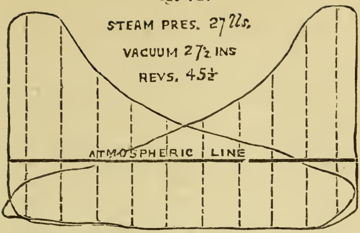


Direct-acting Inclined Paddle-wheel engines.
Indicator diagrams showing the expansion by the link motion. Original scale, $\frac{1}{16}$ in. = 1 lb. Half size.

Fig. 70 were taken from the same engines, and indicate the result of using the link motion and slide-valve for the early cut-off of the steam.

The diagrams that are illustrated by Fig. 71 are superior to any we have yet depicted; for the supply-steam, expansion, exhaustion, and lead lines are perfectly formed, as well as being duplicates; indeed, so far are they worthy of appreciation that we solicited from Mr. Penn a sketch of the indicator he used at the time they were taken, to enable the entire matter to be fully understood—which we illustrate by Fig. 72.

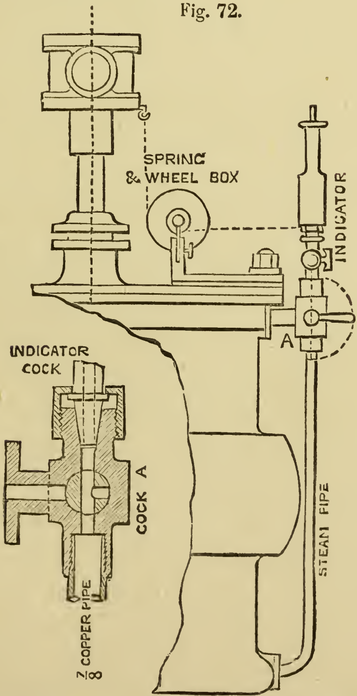
Fig. 71.



OSCILLATING PADDLE-WHEEL ENGINES.

Perfect indicator diagrams taken from the engines fitted in Mr. Penn's steam-yacht "Lara." Gridiron expansion valves on steam-pipes. Diameter of cylinder, 30 in.; stroke of piston, 2 ft. 9 in.; nominal horse-power collectively, 50. Original scale, $\frac{1}{16}$ in. = 1 lb. Half size.

Fig. 72.



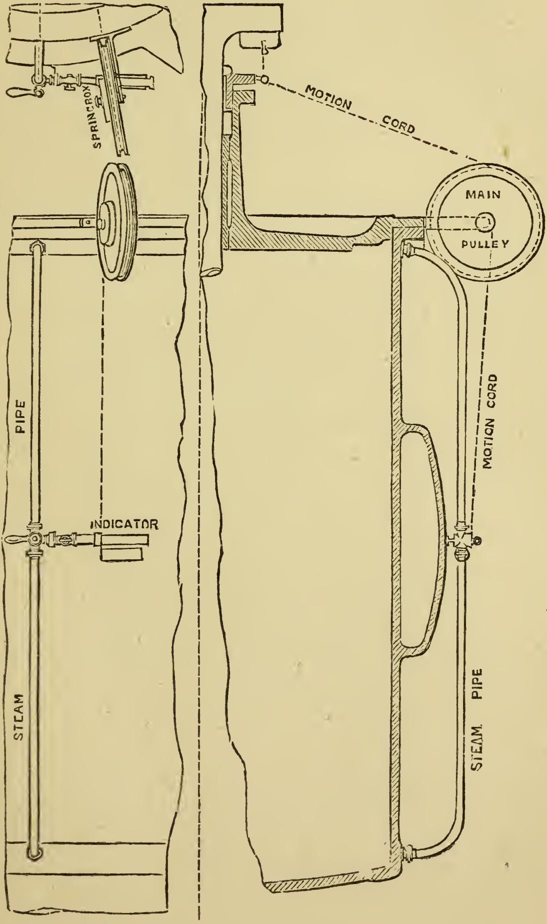
Messrs. Penn's indicator gear for oscillating engines. Scale, $\frac{5}{8}$ in. = 1 ft.

Messrs. Penn's indicator gear for oscillating engines.—This arrangement shows that the indicator is vertically situated on the top of the steam-pipe, and below the instrument an angular two-way cock forms the connection with the pipe; we have shown the section of this cock, union joint, and pipe under the cylinder cover. The arrangement of the reducing motion gear is a spring wheel and a series of spur wheels and pinions, contained in a box of suitable form and dimensions: this is found to be more reliable than the large wheel and spring-box.

Messrs. Napier's indicator gear for oscillating engines.—This "gear," as shown by Fig. 73, is very like that illustrated in pages 22 and 24, excepting that in this case the pipes are attached to the *sides* of the cylinder, instead of the ends. The indicator also is horizontally situated, and the bracket for supporting the main motion wheel is shorter. The plan and front view of the arrangement is shown too, so that the matter is clear without description. Our reason for illustrating this gear in this portion of the present work is that, as the diagrams shown by Figs. 56 and 57 in page 121 are such exceptionally good productions, we solicited a drawing of the indicator and gear from the firm to show

to our readers the arrangement that produced such good results.

Fig. 73.



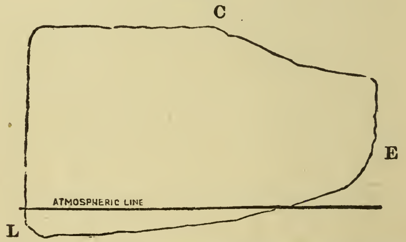
Messrs. Napier's indicator gear for oscillating engines. Scale, $\frac{1}{2}$ in. = 1 ft.

CHAPTER VIII.

EXAMPLES OF INDICATOR DIAGRAMS TAKEN FROM LAND ENGINES.

COMPOUND BEAM-ENGINES.—We commence this chapter with an example of diagrams from a compound beam-engine constructed by a well-known firm in London for a seaport water-works. Fig. 74 is the high-pressure diagram, which

Fig. 74.



COMPOUND HIGH AND LOW PRESSURE STEAM-CONDENSING BEAM ENGINE.

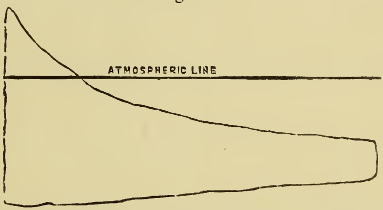
High pressure indicator diagram. Original scale, $\frac{1}{20}$ in. = 1 lb. Half size.

shows that the point of cut-off, C, is about five-eighths of the stroke of the piston. The portion of the exhaust line at E indicates the full release of the steam from the high-pressure into

the low-pressure cylinder. The lead-corner L shows the point of readmission.

The low-pressure diagram is depicted by Fig. 75, which shows an almost perfect expansion curve, correct exhaust, and the least amount of lead necessary.

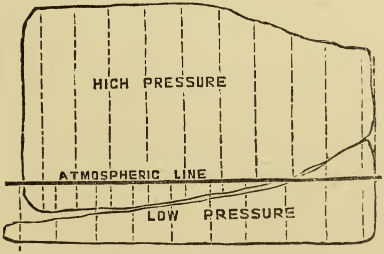
Fig. 75.



COMPOUND HIGH AND LOW PRESSURE STEAM-CONDENSING BEAM-ENGINE.

Low pressure indicator diagram. Original scale, $\frac{1}{10}$ in. = 1 lb. Half size.

Fig. 76.



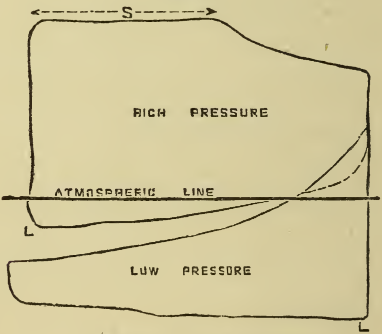
COMPOUND BEAM-ENGINE.

The high and low pressure indicator diagrams pieced together at the *same* scale in lbs. High pressure diagram, half size; low pressure diagram, quarter size in height, and half size in length.

The diagrams illustrated by Fig. 76 depict

what *must* be done to *faithfully* "piece" the figures together, and show also that the valves were well arranged to cause such a perfect connection; as the amalgamation of the two diagrams depicts the action of the steam in both of the cylinders as a combination: also the action of the same volume of steam in two cylinders on opposite sides of the relative pistons. The diagrams shown by Fig. 77 are introduced to further show to the

Fig. 77.



COMPOUND BEAM-ENGINE.

The high and low pressure indicator diagrams of *unequal* scales in lbs. "pieced" together. Original scale, high pressure, $\frac{1}{20}$ in. and low pressure, $\frac{1}{10}$ in. = 1 lb. Half size.

uninitiated that the pressure of the steam regulates the forms of the diagrams. These were taken from the same engines as those preceding, and depict that the "piecing" together at separate scales does not show the actual result.

Horizontal condensing direct-acting engines.—We have only one example from this class of engine, as depicted by Fig. 78, which

Fig. 78.

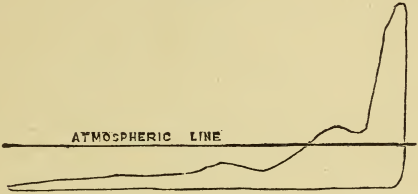


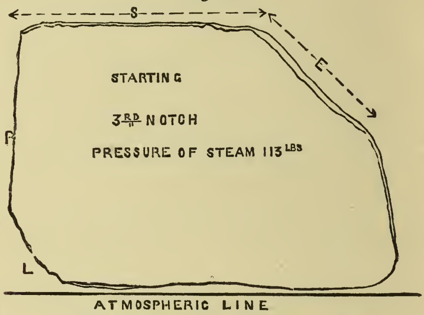
Diagram taken from the "Allen" engine at the French International Exhibition of 1867. Original scale, 1 in. = 24 lbs. Half size.

shows a high degree of expansion, and an early cut-off to produce it with perfect exhaustion and lead. The undulations of the expansion line result from the slackness of the motion-cord and the jerking of the paper-barrel.

Locomotive engines.—We cannot commence this section more appropriately than by illustrating two indicator diagrams taken when the engine started, as shown by Fig. 79. These are creditable examples, as they depict full-supply steam, S, correct expansion, E, good exhaustion with very little back-pressure, ample lead, L, and a vertical admission line, R. A greater length of supply line, S, and a shorter exhaustion line,

E, is shown by Fig. 80, and also the proportionate amount of exhaust line and lead curve, L, with the admission line, R.

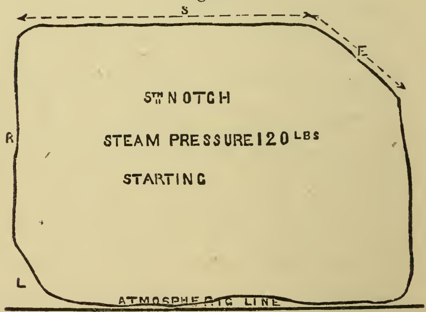
Fig. 79.



LOCOMOTIVE ENGINE L.S.W.R.

Indicator diagrams from the engine when starting, showing full steam. Original scale, $\frac{1}{40}$ in. = 1 lb. Half size.

Fig. 80.

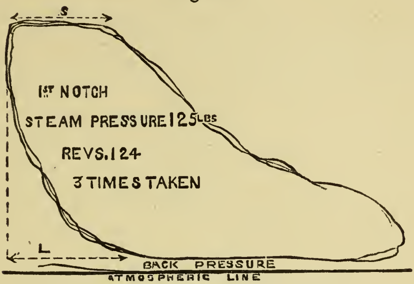


LOCOMOTIVE ENGINE L.S.W.R.

Indicator diagram showing the limit of *full* steam on starting. Original scale, $\frac{1}{40}$ in. = 1 lb. Half size.

Fig. 81 is an example of three diagrams taken

Fig. 81.



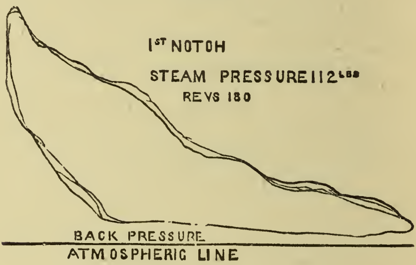
LOCOMOTIVE ENGINE L.S.W.R.
Indicator diagrams showing the proportion of time to velocity. Original scale, $\frac{1}{40}$ in. = 1 lb. Half size.

when the cranks were revolving 124 times per minute. The proportion of the line S to the line L clearly proves that what is taken from the *full* supply line is put on to the lead line; as indeed it must be, because the *lead portion of the crank-pin's circle is opposite to that for supply*; therefore the *time* for each operation is relative, because the crank-pin moves at the same speed above and below the plane line of motion.

The diagrams shown by Fig. 82 present a greater contrast, and Fig. 83 the greatest, with the same amount of back-pressure; which illustrates again that *time* regulates the indication as much as the pressure. The diagrams depicted by Fig. 84 illustrate a further explanation of the pro-

portion of time to speed of the piston, proving, too, that the pressure of the steam is more perfectly indicated with high speeds than the other portions of the matter.

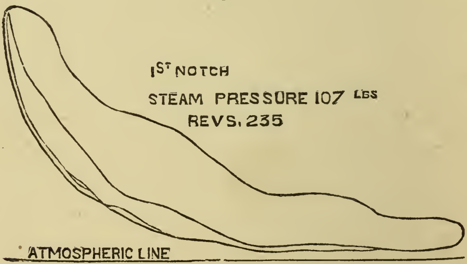
Fig. 82.



LOCOMOTIVE ENGINE L.S.W.R.

Indicator diagrams showing a greater indication of the proportion of time to velocity. Original scale, $\frac{1}{40}$ in. = 1 lb. Half size.

Fig. 83.



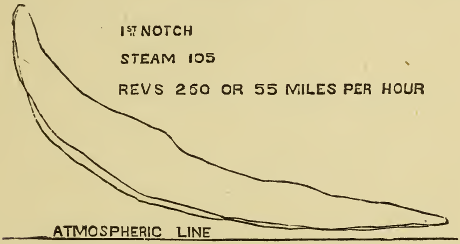
LOCOMOTIVE ENGINE L.S.W.R.

Indicator diagrams showing a high rate of expansion and enlarged lead. Original scale, $\frac{1}{40}$ in. = 1 lb. Half size.

These diagrams, Figs. 79 to 84 inclusive, just referred to, were kindly contributed by C. F. T.

Young, Esq., C.E., and were taken by him from the cylinders of the locomotive "Eagle," on the London and South-Western Railway, during journeys from Waterloo to Southampton and return

Fig. 84.



Locomotive Engine L.S.W.R.
Indicator diagrams showing the result of express speed at the rate of 55 miles per hour. Original scale, $\frac{1}{40}$ in. = 1 lb. Half size.

—a distance, one way, of 161·5 miles. The train left London at 11 A.M. and arrived at Southampton at 3 P.M. On leaving Waterloo the number of carriages was 20, and from Southampton, 16, and others were taken on and off at various points of the line, the average load being 13·11.

The cylinders of the engine are outside, and are 16 in. diameter, with a 22-inch stroke of piston; outside lap of the slide-valve, $\frac{7}{8}$ in.; lead, $\frac{3}{16}$ in.; working pressure of steam, 120 lbs. on the square inch in the cylinders; coupled driving wheels 6 ft. in diameter.

CHAPTER IX.

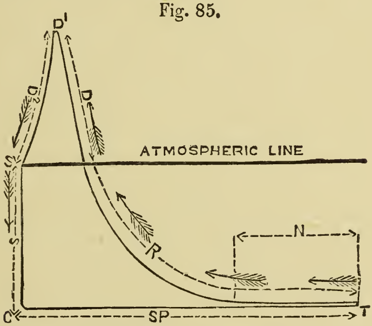
EXAMPLES OF INDICATOR DIAGRAMS TAKEN FROM AIR AND WATER PUMPS.

Explanation.—The principles on which the pump indicator diagram is founded are precisely as those from the steam-cylinder, but the mechanical construction is entirely opposite. The cause for this is that the motion for the pencil when illustrating the pressure of the water is *rising* during the motion of the paper-barrel; whereas with steam it is stationary. Now to render this matter clear to the student, let us imagine a pump with the piston at one end, and the barrel *filled* with water; the action of the piston on moving is to compress the water until the discharge valves open, when the water will be driven out with a velocity due to that of the piston. We will again assume that the same piston is at the end of its stroke, but the barrel only *partially* filled with water; the result will be that on the piston moving for a certain distance there will be *no* pressure, because the water is being merely pushed, and maintaining its level due to its

gravity during that time; but immediately the water fills the space in front of the piston, then the pressure occurs, because the water is lifted above its natural level.

As a practical illustration of this we introduce the diagram Fig. 85.

Fig. 85.



AIR AND WATER PUMP INDICATOR DIAGRAM.

S = Starting point.
s = Vacuum limit.
S P = Stroke of piston.
N = No pressure.
R = Return stroke.
D = Compression and commencement of discharge.
D^1 = Full discharge.
D^2 = Termination of discharge.
C = Commencement of stroke.
T = Termination of stroke.

Original scale, $\frac{1}{8}$ in. = 1 lb. Half size.

The pencil described this diagram under the following circumstances. First, the paper vibrated and the atmospheric line was described. The communication from the air pump was then formed by turning the stop-plug; the pencil then instantly fell for the length of *s*, because

the piston of the pump formed a *suction* directly it moved, which caused the atmosphere to force the indicator piston down to the point of resistance. The pencil here became stationary, and was kept in that position—by the atmospheric pressure—during the stroke of the pump-piston, S P, or the length from C to T. This piston then returned and shifted the water directly in front of it in the barrel for the length of N, which depicts that during that portion of the stroke there was but little, if any, pressure in the pump. The pencil having traced the parallel line, or nearly so, for the length of N, it rose and formed the remainder of the line R, while the pump-piston moved, until it reached the atmospheric line.

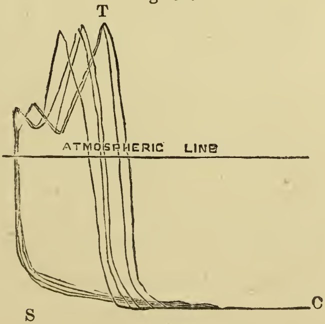
It is perhaps necessary, for further explanation, to state that while the pencil formed the line R the water in the pump—in front of the piston—was constantly maintaining its *natural* level; but as there was some *air* amongst the water, that vapour rose as the liquid was disturbed, and thus caused the pressure as indicated *below* the atmospheric line.

The compression and discharge of the water here commenced, and thus the *load* on the piston caused a further pressure under the indicator piston

which forced the pencil to rise for the height of D. Here it stopped, and the pressure maintained its level for a moment, which allowed the pencil to form D^1. The main load on the piston was here overcome because the bulk of the water left was of a lesser quantity than what had been discharged; then, as the pressure decreased so did the pencil descend, and form the line D^2, and then resume its original position at S. We have introduced the arrows to indicate the motion of the pencil, so that the student can fully understand this subject.

By the next example, shown by Fig. 86, we

Fig. 86.



Double-acting Air-pump; Horizontal Motion; Screw-engines. Messrs. Watt's indicator diagrams, showing sudden discharge and a fair attainment of vacuum. Original scale, $\frac{1}{8}$ in. = 1 lb. Half size.

illustrate that the pencil started from C; and as there was *no* pressure for nearly half the stroke

of the pump, the lines parallel with the atmospheric line were described, and that as the velocity of the piston was 168 strokes per minute, the pencil rose almost vertically to record the limit of the discharge pressure, T, and nearly as quickly fell when the release occurred. The vacuum attained was 26 in., and the pressure of discharge $11\frac{1}{2}$ lbs., being $1\frac{1}{2}$ lb. less than the vacuum.

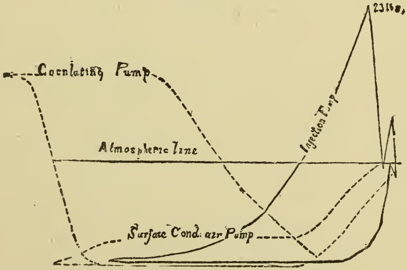
The undulations made by the pencil on descending above the atmospheric line are produced by air being mixed with the water, and thus there was an unequal pressure at the final discharge. The pencil's descent to the vacuum line was perpendicular for nearly that distance, the curved portion S showing also that the pencil did not descend as rapidly as the pump's piston moved, and that the full suction occurred and was maintained when the half-stroke was made.

The diagrams from vertical pumps next come under consideration, and illustrations of them are given by Fig. 87. The injection-pump diagram shows a rapid discharge up to a pressure of 23 lbs. from the vacuum line.

The circulating pump diagram indicates a continuous discharge, and that there was but little clearance for the water to " wash " in the barrel before the pressure occurred, as the angular or

pressure line commences at the beginning of the diagram, and thus shows that the water began to leave the barrel at about one-third of the stroke of the piston instead of two-thirds, as in the injection-pump. The cause of the discharge limit line being horizontal for such a length is, that the piston had to force the water through the tubes of the condenser, and thus the resistance was continuous during that process.

Fig. 87.



DOUBLE-ACTING PUMPS; VERTICAL MOTION; SCREW-ENGINES.
Mr. Spencer's indicator diagrams, showing the difference in the discharge pressures. Scale, $\frac{1}{16}$ in. = 1 lb.

The surface-condenser air-pump diagram indicates below the atmospheric line, and shows that the discharge pressure was never above the atmosphere, and that at the return-stroke there was a little pressure in the barrel, as indicated by the angular line.

CHAPTER X.

THE FORMULÆ REQUISITE TO DEFINE THE DUTY OF AN ENGINE FROM THE INDICATOR DIAGRAM.

We have endeavoured to "drill" the student in the preceding chapters, and we now give him our concluding instructions as to the use of his "attention" for practical purposes.

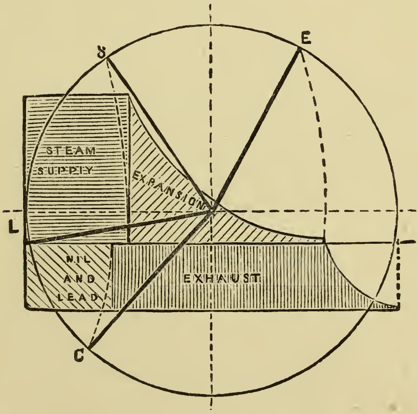
To commence with, we must notice the difference that must occur in indicator diagrams taken from different classes of engines. Now, with engines of slow *rotative* velocity, the diagram taken will always be more truthful than with a high revolving motion for the crank-shaft and a short stroke for the piston, because there is *more time* in the former case than in the latter for the steam to expand and exhaust; and, as a proof of this, it will have been noticed doubtless that the expansion and exhaustion lines in most of the diagrams we have illustrated are the most faulty portions of the whole.

Of course the real theoretical line for expansion should commence from the supply line and terminate at the atmospheric line; and this can

best occur with engines having a long stroke of piston, an early cut-off, and cam-motion for the working of the equilibrium circular valves that are separately situated at each end of the cylinder.

When the "slide"-valve regulates the admission, expansion, and exhaustion of the steam, the perfect indicator diagram will be as illustrated by Fig. 88, which shows full steam, per-

Fig. 88.



A perfect indicator diagram from a cylinder fitted with a slide-valve and direct-acting link motion in full gear; also showing the position of the piston and crank-pin at each limit of the progressive construction of the diagram.

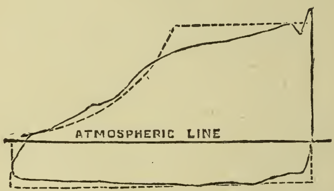
fect expansion, quick exhaustion, and continuous vacuum, with the least amount of lead.

The position of the crank-pin when the slide-valve is cutting off is at S; when it is at E the

valve is permitting exhaustion, and that continues until the crank-pin reaches C, usually termed the "compression" point; from here there is supposed to be no admission or escape until the crank-pin reaches L, the lead position, when the readmission occurs.

The diagram Fig. 89 is introduced as a con-

Fig. 89.



An indicator diagram showing an inclined supply-steam line, with full expansion, exhaustion, and undulated lead.

trast to that preceding, which illustrates the imperfections of the gear used for the supply of the steam; if the pressure in the boiler were constant and the gear perfect the dotted lines indicate the formation that should have occurred.

We will now suppose that the student has taken an indicator diagram, and he desires to prove its utility, and as an illustration for the purpose we introduce the Fig. 90.

The first process is to divide the length of the figure into *ten* equal divisions, and then to set off

half of one space from each extremity and arrange the remainder equidistant. Next, lines are drawn at right angles with the atmospheric line from each intersection or divisional point; then, with

Fig. 90.

The correct method of obtaining the mean pressure of the steam and vacuum as recorded by an indicator diagram.

the scale of the diagram, each ordinate is measured and figured above and below the atmospheric line; the total sum of each set of ordinates is then obtained by simply adding the several lengths together, and the mean is known by dividing the total by the number of the ordinates.

In the example, Fig. 90, the mean pressure of the steam is 15·575 lbs. on the square inch; and

the vacuum obtained is equal to an atmospheric pressure exhausted of 9·69 lbs. on the square inch only. Observe, that the vacuum is recorded as "v. 26," or 13 lbs. pressure by the gauge, and the steam as $27\frac{1}{2}$ lbs. in the boiler; whereas in the cylinder the highest record of full supply is only 24·2 lbs. This reduction of the steam pressure is due to the friction and radiation; and the difference in the vacuum pressure is owing to the temperature being higher in the cylinder than in the exhaust steam pipe at the condenser end.

Our next step is the use of the mean pressure known; the cylinder is $46\frac{9}{16}$ inches in diameter; the area being known from the square of the diameter $\times$ ·7854 = 1792·7994 square inches; then 1792·7994 $\times$ 25·265 (the mean pressure of both steam and vacuum) = 43021·2268 lbs. as the total load.

The length of the stroke of the piston is 1 ft. 6 in., and the speed is known by the formula—number of revolutions of the crank-shaft $\times$ twice the length of the stroke of the piston.

Next, 1·5 $\times$ 2 = 3 feet, and as the number of revolutions is recorded as R. 116 per minute, the speed of the piston = 116 $\times$ 3 = 348; then, 348 $\times$ 43021·2268 = 14971386·9264 = the total

power in foot-pounds, because the speed is taken in feet.

The sum of 33,000 is the constant to equal one horse-power, or the weight in pounds that a horse can lift one foot high per minute.

Then, if the total power is divided by the constant, the quotient equals the indicated horse-power; for example, the total power $\frac{14971386 \cdot 9624}{33000} = 453 \cdot 6783$ horse-power of the engine under notice; and as the nominal horse-power is 100, the relation of the indicated power to the nominal is obvious.

The formula, therefore, for the indicated power of an engine is thus: when

$$\frac{M \times A \times [2 \times S \times R]}{33000}$$

M = Mean indicated pressure in lbs. per square inch.

A = effective area of the cylinder in square inches.

S = Stroke of the piston in feet.

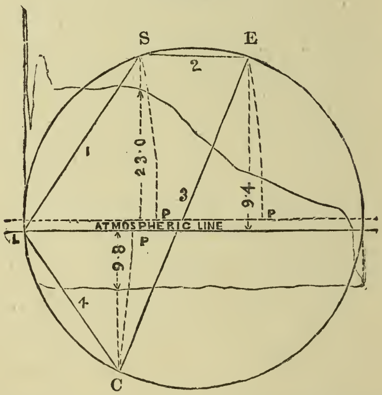
R = Number of revolutions per minute.

Of course with high-pressure engines the mean pressure is obtained from the mean of the depth of the diagram above the atmospheric line.

As the points on the diagram where the limits

of the duty of the steam occurs are both instructive and interesting, we introduce the Fig. 91, which is a repetition of the diagram, Fig. 90, in page 151, which being figured as this one at the several pressures, it fully depicts their relation. The point S on the circle indicates that when the

Fig. 91.



A diagram showing the positions of the crank-pin during certain stages of the construction of the indicator diagram, Fig. 90 in page 151.

crank-pin reached it, supply steam terminated; at E the exhaustion of the steam commenced; at C the steam-ports were closed on that side of the piston, and at L the lead commenced; therefore No. 1 is the chord of supply, No. 2 of expansion, No. 3 of exhaustion, and No. 4 of nil or compression; and the points P P P are the relative positions

of the piston to the crank-pin. This method is precisely as that illustrated by the frontispiece plate, the geometry of which originated from the author, and induced him to write the present work on the INDICATOR DIAGRAM.

INDEX OF SUBJECTS.

INDEX OF ILLUSTRATIONS.

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